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DIGEST OF THE EXPERIMENTS MADE BY MR. MICHAEL LONGRIDGE, CHIEF ENGINEER OF "THE ENGINE, BOILER, AND EMPLOYERS' LIABILITY INSURANCE COMPANY" OF MANCHESTER, ENGLAND, ON THE TRIPLE EXPANSION PUMPING ENGINES AT THE EAST LONDON WATER WORKS, LEA BRIDGE, AND DESCRIBED IN HIS REPORT TO THE COMPANY FOR 1893.

BY CHIEF ENGINEER ISHERWOOD, U. S. NAVY.

Mr. Michael Longridge, the Chief Engineer of "The Engine, Boiler, and Employers' Liability Insurance Company" of Manchester, England, makes regularly an annual report to his company, in which, among much other interesting professional matter, he describes some of the principal engineering experiments made by him during the current year, in the course of his service as expert or as insurer.

As these reports are printed in very limited numbers simply for distribution to the members of the company, they are not easily accessible to the general public, and as the report for 1893 contains the description of a remarkably valuable experiment

made simultaneously on duplicate, direct acting, triple expansion, condensing, steam jacketed, pumping engines, the writer believes he will do his profession a service by preparing a digest of it for the readers of this JOURNAL.

The data of the experiment are given with unusual fullness, and this completeness, taken in connection with the extreme accuracy with which the data were determined, adds enormously to the importance of the results.

The weight of the water of steam liquefaction drawn from the steam jackets of each of the three cylinders, was separately ascertained for each cylinder. The weight of the water of steam liquefaction drawn from the steam pipe was similarly ascertained. The heat radiated per hour from the exterior surfaces of the jackets of the steam cylinders was also ascertained, and separately for each cylinder. The capacity of the receiver between the small and intermediate cylinders, and between the intermediate and large cylinders, is given; together with the clearance surface in each of the three cylinders, and with the steam jacketed surface of each cylinder.

Each steam jacket was fitted at its highest point with a cock for the discharge of whatever air the jacket might contain at the commencement of the experiment; and with a pressure gauge showing the steam pressure in the jacket during the experiment. So far as the writer's knowledge extends, this is the first time that such a method has been followed, yet its necessity is evident, for no reliable experiment can be made on the effects of steam jacketing without it. Evidently, the air in the jackets at the commencement of an experiment cannot be expelled in any other way than by blowing them through with steam, and if this air remains in them, then just as evidently the steam jacketing must fall short of its possible and proper economic efficiency. Again there is no certainty of what the pressure in the jackets is (except in the case of some arrangements in which the steam passes from the steam pipe through the jackets into the valve chests of the cylinders, instead of passing directly into these chests from the pipe) otherwise than by the evidence of a pressure gauge.

With triple expansion marine engines having their intermediate and large cylinders jacketed with steam of about the initial pressure in those cylinders, and taken from the main steam pipe through reducing valves, the jackets are furnished with safety valves loaded to the intended pressures, so as to prevent the possibility of accident by either crushing the cylinder or exploding the jacket were full boiler pressure present. In such a case the jacket safety valves can be made to combine the functions of air discharge cock and of pressure gauge. Such an arrangement, however, is never used when the jackets of the cylinders are supplied with steam of presumably full boiler pressure. In that case, whether the air has been discharged from, or what may be the pressure in, the jackets, is a mere matter of inference, not of knowledge.

In many experiments on steam jacketed cylinders of all types of engine, using high pressure steam with large measures of expansion, the water of liquefaction drawn from the jackets has been, comparatively, quite insignificant in quantity; in fact, scarcely more than was due to external radiation of heat. These experiments have passed unchallenged as regards accuracy, and possibly they were accurate as far as the measured quantities were concerned; but the jackets themselves were faulty either in being largely filled with cores, when cast in one piece with the cylinder, or in leaky joints if the two were cast separately and bolted together, or were not filled with steam of the intended pressure, owing to too small area and too great length of the pipes supplying them with steam. The facts are certain that not less than 8 per centum of the feed water will be drawn from the steam jackets of the cylinders of the largest engines using greatly expanded steam of high pressure, nor less than 16 per centum of the feed water when drawn from the jackets of the cylinders of small engines working under the same conditions; these percentages being the aggregate for all the cylinders of the engine.

A test made of the joints of a steam jacket with the engine at rest, proves nothing as to their tightness when the engine is working. These joints, though tight in the former case, may leak much in the latter case. When the engine is at rest, the

temperature of the cylinder and the jacket is the same, that is to say, the two walls of the jacket have the same temperature when the jacket is filled with steam and the engine is at rest. But as soon as the engine works, this equality ceases, the inner wall or cylinder then having a much lower temperature than the outer or jacket wall, which latter consequently expands more per inch of diameter of jacket than the former does per inch of diameter of cylinder, and there is developed as the result an annular gap between the metals of the jacket and of the cylinder, through which the steam in the jacket leaks into the cylinder, as the pressure in the latter is less than in the former. When the enormous cylinder liquefaction of steam is considered, and the proportional lowering of the temperature of the cylinder metal to cause that liquefaction, there can be easily understood how large must be the difference when the engine is working with greatly expanded high pressure steam, between the temperature of the cylinder metal and that of the outer jacket metal, which last temperature is sensibly constant and nearly the same as the temperature of the steam in the jacket. Further, when the length of the annular gap (which is equal to the outer circumference of the cylinder) is considered, and the difference between the pressures in the jacket and in the cylinder, a width of gap of less than a hair's breadth will be found sufficient to leak a considerable weight of steam per unit of time. The difference of temperature between the jacket metal and the cylinder metal, in a working engine, produces also an unequal elongation of these metals in the direction of the axis of the cylinder, tending to develop leaks between them wherever the design permits.

The discordant economic results from different steam engines, which, apparently, should give accordant results, are often due to mere unsuspected construction defects, producing accidental and unequal leakages of steam or other abnormal consequences; comparison, therefore, must not be depended on between isolated cases, but between averages for groups, and the more numerous the cases constituting a group, the more reliable will be these averages. The greatest caution and discrimination are required, combined with practical experience and theoretical knowledge,

in ascertaining the true underlying general laws, which, in steam engineering, are overlaid, more or less, by the influence of extraneous causes, accidental conditions, qualities of material, faults of design and execution, and the whole chapter of annoying and uncertain inaccuracies, which, though suspected, cannot be detected, and vitiate to a greater or less extent, the experimental results.

Incompleteness of indispensable data which, by compelling the substitution of imaginary inferences for measured quantities, is a frequent misleader in attempts to determine a true generality. The importance of completeness of data cannot be overestimated; for want of it the great mass of engineering experiments are reduced enormously in value. A commencement may be made by accepting hypotheses, but the end must be reached with everything hypothetical rejected. Every fact must be experimentally ascertained and clearly described, nothing being included that has not been either counted, weighed or measured.

No one is more aware than the writer of the extreme difficulty of obtaining completeness of data in the case of experiments on large steam machinery, owing to the expense, time and labor involved. The difficulty is even greater than that of obtaining accuracy of such data as are observed. Moreover, in such cases, the difficulty of ascertaining whether the experimental mechanism is in good condition proper for the experimental purpose, is almost insurmountable. It is simply assumed to be so, but the uncertainty is distressing, and always justifies much doubt in the results, at least quantitively. Hence, the only safety is in basing conclusions on the average results of groups of experiments, each group consisting of machinery so allied in type, details, dimensions, principles and conditions, that consistent results *ought* to have been given, the correction of errors being relied on for neutralizing the numerical differences.

These ideas can best be realized by experiments on pumping engines, which for the purpose of ascertaining the practical conditions that influence, and the extent to which they do influence, the economic production of power by steam used in practical steam engines, are better adapted than engines employed in other

work. Experiments on this class of engines, therefore, are of the highest value, their results being much more certain and free from error of observation or measurement than any other class.

For instance, in the important particular of the determination of the power developed by the engine, if the pressure of the water, or the "head of water" against which the engine works, be constant, then the pressure on the steam pistons will be constant also, let the speed of these pistons be what it may. One indicator diagram, therefore, if its accuracy be certain, is sufficient to determine in connection with the average speed of the piston, the average power developed by the engine during the experiment.

Now, if the steam pressure in the boiler be maintained constant, the speed of the piston will remain constant too, for, with opposing constant water pressure, the speed of the piston is only alterable by alterations in the boiler pressure, other things, of course, remaining the same. If the boiler pressure be increased, the piston speed will be increased, but the indicator diagram, that is to say the pressure on the piston, will remain exactly the same. Likewise, if the boiler pressure be decreased, the piston speed will be correspondingly decreased, but the indicator diagram will remain exactly the same as before, the lower limit of the boiler pressure being reached when that pressure becomes too small to equilibrate the statical water pressure plus the friction of the moving parts of the mechanism, *per se*, plus the resistances of the pump valves, plus the friction resistances of the various surfaces in contact with the moving water, and plus the friction of the load when the pumping power is passed through a rotating or vibrating mechanism, in which case the engine comes to rest, for in order to produce motion, the piston pressure must equal the water pressure added to the pressures equilibrating the above resistances, but after this equality is obtained, no more pressure can be produced upon the piston by increase of boiler pressure, such increase of pressure having for its effect only increased velocity of piston. Thus, with the boiler pressure infinitesimally greater than the sum of the statical water pressure and of the pressures equilibrating the above resistances, the piston will move

with an infinitesimal velocity, which will increase, *pari passu*, as the boiler pressure is increased, the piston pressure remaining constant throughout all the changes of piston velocity, so that the power developed will be directly proportional to the velocity of the piston, and the work done will also be directly proportional to that velocity. Hence, the work done in pumping will be directly proportional to the power exerted, and the economy of the performance will not be affected by the velocity with which the work was done.

None of the foregoing named dynamical resistances in addition to the statical resistance of the water pressure, is affected by the velocity of the piston. They are all, except the resistance by the pump valves, simply frictional resistances uninfluenced by the velocity with which they are overcome. The resistance of the pump valves is also constant at all velocities of piston. Hence, the indicator diagram from the steam cylinder is also constant at all velocities of piston, the mean pressure from that diagram equilibrating the aggregate of all the resistances, represented by pressures, of whatever kind soever.

As an increased boiler pressure does not appear upon the piston, how does it operate to increase the piston speed? Simply by furnishing the steam from the boiler to the piston at the increased rate of the piston speed.

The steam is conveyed from the boiler to the cylinder partly by the push of the boiler pressure, and partly by the expansion of the steam itself, which takes place mainly within the steam pipe and cylinder; the conveying force being the difference between the pressure in the boiler and the pressure in the cylinder previous to the point of cutting off. There is a continuous decrease of pressure from the boiler heating surface generating the steam to the point in the cylinder at which the steam is cut off. The decrease is scarcely sensible in the boiler, owing to the large capacity of the steam room proportionally to the cross area of the steam pipe; but from the boiler end of the steam pipe to the point in the cylinder at which the steam is cut off, the fall of pressure and consequent expansion of the steam becomes more or less strongly marked according to circum-

stances. In fact, the steam is always used with some measure of expansion on the piston previous to being cut off by the closing of the expansion valve. The universal belief is, that the steam is worked expansively on the piston only after the cut off valve has closed. This is an entire mistake; it is always worked expansively throughout the entire stroke of the piston, before as well as after the closing of the cut off valve. Sometimes the expansive working prior to the closing of the cut off valve is very small, sometimes quite large, according to conditions, but it is always something. The only means by which the boiler pressure can be produced in the steam pipe, and in the cylinder previous to the closing of the cut off valve, is by backing against the piston an opposing resistance equal to the boiler pressure; that is to say, the boiler pressure on one side of the piston is to be balanced by an equivalent resistance against the opposite side of the piston, in which case there could not be any movement of the piston, or power developed. With any less opposed resistance, there must necessarily be a continuous decrease of the boiler pressure as described, and if the pressure is decreased, there must have been an accompanying corresponding expansion of the steam. Now, with an engine working with any given velocity of piston against a constant resistance, that is a resistance uninfluenced by the velocity with which it is overcome, as in the case of a pumping engine pumping against a constant head of water, the load due to which offers a constant resistance at all speeds of piston, the supply of steam from the boiler is just sufficient to maintain the given velocity of piston, the boiler pressure remaining constant. The piston velocity in this case cannot be increased, because the boiler cannot furnish more steam per unit of time to equilibrate the resistance moving at a higher speed; the steam supply and the work done per unit of time being exactly balanced. What is required, and it is all that is required, to increase the piston speed, is a delivery of more steam per unit of time upon the piston, *but of the same piston pressure as before*. This can be effected by simply increasing the boiler pressure, the initial pressure upon the piston remaining unchanged; or, in other words, by increasing the *head of steam* in order to

produce a more rapid flow of steam from boiler to cylinder; the velocity of the flow having a definite relation to the difference between the pressure in the boiler and in the cylinder, the latter pressure remaining constant, or, in other words to the head of steam employed. Of course, other things being equal, the greater this head of steam the more rapid will be the piston speed, the consumption of steam per unit of time being in the direct ratio of the piston speed, as will also be the power developed and the work done, the economy of the steam remaining unchanged at all speeds of piston. The ratio of the useful to the prejudicial work done, will also remain constant at all speeds of piston.

The facts being experimentally indisputable that the resistance to a *perfectly smooth* plane surface moving through an indefinite extent of water, in the direction of the surface, is *nearly, but a little less than*, in the ratio of the square of the velocity of the surface, the power to produce different velocities being in the ratio of the cube of the velocity; while, in the case of a pumping engine, the resistance offered to the moving water pumped, by the surfaces of the mains or of the receiving and delivering pipes, is constant, that is, not affected by the velocity, requires an explanation.

The two phenomena are not only entirely distinct, but are of essentially different kinds. In the first case, that of the plane surface moving in an indefinite extent of water, in the direction of the surface, the resistance is due to the adhesion of the watery molecules to the surface, and to the cohesion of these molecules among themselves. The adhesion of the molecules to the moving surface, produces a movement of the layer of them in contact with the surface, in the direction of the surface, and the cohesion of the molecules to each other transmits this movement from the first layer to the next, and from that to its next, and so on, the velocity of each successive layer decreasing until the original force is sensibly spent. The first layer in contact with the surface has not the velocity of the surface but a much less velocity, as the adhesion is overcome by the weight and resistance of the successive layers of watery molecules attached to the first layer by cohesion, so that the first layer is continuously being dislodged from

the surface by sliding along the surface in the reverse direction of the movement of the latter. The resistance to this sliding is constant, that is, it is not affected by the speed with which the molecules slide. Nor is the adhesion of the water to the surface affected by the pressure of the water, nor is the resistance of this adhesion similar in any way to the resistance of friction between the smooth surfaces of moving solids in contact with each other. Adhesion is a form of attraction between the solid surface and the liquid in contact with it. If there were no adhesion there would be no resistance of the liquid to the movement of the surface in it in the direction of the surface. When a surface is dipped into water and withdrawn, it remains *wetted* though held vertically, and the wetting is the result of an attractive force. The minimum resistance of liquid to surface, is when there is a repulsion between the two. The resistance overcome is consequently composed of the watery adhesion, which is a constant resistance at all speeds; and of the sensible movement of water, which is in the ratio of the square of the speed. Hence results that the resistance to the moving surface as a whole, is a little less than the square of the velocity of the surface, the exact power of the speed being variable with the smoothness of the surface, the rougher the surface the nearer will the law of the squares be approximated. The rougher the surface the greater will be its resistance at equal speeds, the roughnesses acting as projections from the surface into the water, and giving motion to not only a greater quantity of water, but imparting a greater velocity to that greater quantity than in the case of a smooth surface. The resistance in this case is the same as that of an oar projected into water from the side of a boat and held there. Evidently, for equal speeds of surface, as the resistance due to the putting of the water in motion, caused by the roughness of the surface, becomes greater and greater, while the resistance due to overcoming the adhesion of the water remains sensibly constant, the nearer will the aggregate resistance approach the law of the squares, though never actually attaining it. For practical purposes, and with the smoothest practical surfaces, the experimental results so nearly approach the law of the squares that, considering the uncertainty of the numerical co-

efficients, the law of the squares may be adopted as within the limits of experimental errors.

The coefficient of resistance of the unit of surface of plane surfaces of different lengths but equal breadths, moving at equal speed in an indefinite extent of water in the direction of the lengths of the surfaces, varies much from a constant, the longer the surface the less per unit of surface will be the coefficient of resistance. The law of the decrease in function of increase of length, has not been experimentally determined, but the decrease for equal increments of length becomes smaller and smaller as the lengths of the plane become greater and greater; a length is thus reached beyond which the coefficient of resistance per unit of surface decreases so slightly for additional lengths, that, although there must always be some decrease with increase of length, it becomes small enough to be practically negligible and the coefficient may be considered as constant.

A rigorous limitation must here be insisted on as regards the application of the principle, namely, that the law applies to only perfectly *plane* surfaces, and to such curved surfaces as have a uniform curvature crosswise or in the direction of their breadth and for their entire length, but no curvature in the direction of their length; it does not apply to either warped or twisted surfaces, nor to any other kind of curved surface except the one just described; consequently, it cannot be applied either to the external immersed surfaces of vessels, or to the helicoidal surfaces of screw propellers. In selecting the coefficient of resistance for the unit of both these surfaces, they must be considered as of infinitesimal length, and the proper coefficient will, in their cases, be the experimental maximum.

The reason why the coefficient of resistance per unit of the surface of plane surfaces of different lengths, but otherwise the same, moving with the same speed in the direction of their lengths in an indefinite extent of water, decreases as the length increases, is as follows: The speed of the water entrained by the moving surface, is constant at all points of the surface. The mass of this entrained water has a definite and uniform thickness, consequently, it presents an area in the direction of its motion equal

to the product of this thickness into the breadth of the surface, and this area has to be forced against the quiescent water in front of it, displacing that water. In other words, it has a head area to be driven through the surrounding water, which offers a corresponding resistance, and this resistance is the same, let the length of the surface be what it may. Now the number of units of surface in planes of equal breadth but unequal length, is directly as their length, consequently, the head resistance of the mass of water entrained by the planes, will be, per unit of surface less and less as the length of the planes is more and more. This *particular portion*, therefore, of the *aggregate resistance* of water to a plane surface moving lengthwise in it, will diminish, *per unit of surface*, in the direct ratio of the length of the plane surface, the breadth being constant, and will cause the aggregate resistance to be correspondingly reduced per unit of surface. The fact that doubling the length of a plane surface moving lengthwise in water at constant speed does not double the aggregate surface resistance, has been experimentally known for a century, but this is the first time a correct explanation has been given of the cause.

The resistances by the inner surfaces of the pipes or mains through which water is pumped, follow entirely different laws from those which have been described for plane surfaces moving in an indefinite extent of water and in the direction of the surfaces. It is necessary that the laws in both cases should be developed, in order that either may be understood, and that the errors should be avoided which occur by applying to one case what is peculiar to the other.

The resistance of the water—apart from that of its weight—in the case of pumping it through *straight* pipes, is simply the resistance of its adhesion to the inner surface of the pipes, and the resistance due to the roughness of this surface, the less the adhesion of the water and the smoother the surface of the pipes, the less will be the resistance of the water to movement through them. The resistance of the water is also affected by its temperature, the higher the temperature the less will be the resistance; the law of the decrease, and the numerical value of the decrease

are unknown. If the pipes are bent or have curvature, their resistance to the movement of water through them will be increased in the inverse ratio of the radii of the curvature, and in the direct ratio of the arcs of curvature. The resistances of pipes will also be in the direct ratio of their lengths, and in the inverse ratio of their diameters. The quantity of water momentarily in action does not increase with the speed of the pumping but remains constant, whereas the quantity of water momentarily in action over the surface of a plane moving in the water in the direction of that surface with different speeds increases in the ratio of the speed, so that the two cases are subject to widely differing laws.

The water in the pipes does not move with the same velocity in every part of the cross area of the pipe; the greatest velocity is at the axis of the pipe and the least velocity is at its periphery; the velocity increases from the periphery to the axis according to an unknown law; the mean velocity only is known, consequently, the force of the adhesion of the water per unit of surface of the pipe cannot be ascertained from a pumping experiment; such an experiment made with observation of the complete data allows in any case the determination of only the aggregate resistances of the pipe as a whole but not of these resistances separately.

Although in different pumping engines, the ratio of the resistances of the pipes relatively to the weight of water pumped per stroke of pistons, must vary largely, this ratio for the same engine will be constant for all speeds of pumping, so that the same indicator diagram will be obtained from the same steam cylinder let the number of strokes made by its piston per unit of time be what it may, provided the pumps are filled to the same capacity. The weight of water delivered by the pipes per unit of time multiplied by the height through which that water was lifted by the engine, and the product divided by the weight of fuel or of steam expended in the same time, does not give reliable comparisons of the economy of the pumping; such comparisons can only be obtained by dividing the work done per unit of time by the engine as measured by the indicator diagram, by the weight of the fuel or of the steam consumed during the same time. In

other words, the economy of pumping engines, relative or absolute, can only be obtained in the same manner as for other engines. The dynamical resistances overcome by pumping engines is not only greater than the statical resistance, but greater in very different ratios according to the condition of the cases. The method of obtaining the dynamical resistance against which the pumps work, by means of pressure gauges inserted in the pipes beyond the delivery valves of the pumps, gives very uncertain results. The pressure of a fluid at rest can be accurately ascertained, but the pressure of a fluid in motion is a difficult datum to determine with certainty, and the uncertainty increases with the swiftness of the motion.

ENGINES.

The engines are two in number, and are of the triple expansion, vertical, direct acting, surface condensing, steam jacketed, type; they are exact duplicates. Each engine consists of three cylinders, the axes of which are in the same vertical plane, and all have the same stroke of piston, one piston rod, and no tail rod. The crosshead of each cylinder is sufficiently long to have a journal near each end, these journals being far enough apart to allow a connecting rod from each to clear the vertical pump which stands immediately below the cylinder and has its plunger or ram operated directly by an extension of the piston rod of the cylinder. Beneath the pump is the crank shaft, which is revolved by the two connecting rods from the crosshead journals, one rod on each side of the pump. The crankshaft has two flywheels, one at each end. The three crankshafts—one for each cylinder—are coupled together so as to form a single shaft underlying the whole engine, the pairs of cranks being equispaced around the shaft, or placed 120 degrees apart.

The cylinders are provided with Corliss valves, one for steam admission and one for exhaust delivery at each end of each cylinder, the steam valve acting also as cut off valve. Each cylinder is steam jacketed on top, bottom and side.

Each engine has a surface condenser, consisting of a group of iron tubes placed in the culvert through which the water to be

pumped passes on its way to the suction pipe. The exhaust steam is within the tubes, and the refrigerating water is on the outside of them.

The pumps are single acting and vertical, one to each cylinder; they deliver their water directly into the mains.

The following are the principal dimensions and proportions of each engine:

Number of steam cylinders in each engine	3
Diameter of the small cylinder, inches	20
Diameter of the piston rod of the small cylinder, inches	5
Mean effective area of the piston of the small cylinder, square inches	304.3425
Stroke of the piston of the small cylinder, feet	4
Space displacement of the piston of the small cylinder per stroke, cubic feet	8 454
Space in the clearance and steam passages at one end of the small cylinder, cubic foot	0.4
Fraction which the space in the clearance and steam passages at one end of the small cylinder is of the space displacement of its piston per stroke,	0.047
Aggregate jacketed surfaces of the small cylinder, square feet	29 5
Aggregate surface in the clearance and steam passages at one end of the small cylinder, square feet	11.5
Diameter of the intermediate cylinder, inches	34
Diameter of the piston rod of the intermediate cylinder, inches	5
Mean effective area of the piston of the intermediate cylinder, square inches,	898.105
Stroke of the piston of the intermediate cylinder, feet	4
Space displacement of the piston of the intermediate cylinder per stroke, cubic feet	24.947
Space in the clearance and steam passages at one end of the intermediate cylinder, cubic foot	0.97
Fraction which the space in the clearance and steam passages at one end of the intermediate clyinder is of the space displacement of its piston per stroke	0.039
Aggregate jacketed surfaces of the intermediate cylinder, square feet	50.9
Aggregate surface in the clearance and steam passages at one end of the intermediate cylinder, square feet	25.2
Difference between the effective areas of the pistons of the small and intermediate cylinders, square inches	593.762
Diameter of the large cylinder, inches	57
Diameter of the piston rod of the large cylinder, inches	5
Mean effective area of the piston of the large cylinder, square inches	2,541.947
Stroke of the piston of the large cylinder, feet	4
Space displacement of the piston of the large cylinder per stroke, cubic feet,	70.609
Space in the clearance and steam passages at one end of the large cylinder, cubic feet	2.8
Fraction which the space in the clearance and steam passages at one end of the large cylinder is of the space displacement of its piston per stroke,	0.039

Aggregate jacketed surfaces of the large cylinder, square feet	88.0
Aggregate surface in the clearance and steam passages at one end of the large cylinder, square feet	63.8
Difference between the effective areas of the pistons of the intermediate and large cylinders, square inches	1,643.842
Ratio of the effective area of the piston of the large cylinder to the effective area of the piston of the small cylinder	2.951
Ratio of the effective area of the piston of the intermediate cylinder to the effective area of the piston of the small cylinder	8.352
Ratio of the effective area of the piston of the large cylinder to the effective area of the piston of the intermediate cylinder	2.830
Ratio of the aggregate space displacement per stroke of the piston of the intermediate cylinder and the space in the clearance and steam passages of that cylinder, to the aggregate space displacement per stroke of the piston of the small cylinder and the space in the clearance and steam passages of that cylinder	2.927
Ratio of the aggregate space displacement per stroke of the piston of the large cylinder and the space in the clearance and steam passages of that cylinder, to the aggregate space displacement per stroke of the piston of the small cylinder and the space in the clearance and steam passages of that cylinder	8.291
Ratio of the aggregate space displacement per stroke of the piston of the large cylinder and the space in the clearance and steam passages of that cylinder, to the aggregate space displacement per stroke of the piston of the intermediate cylinder and the space in the clearance and steam passages of that cylinder	2.832
Capacity of the receiver between the small and the intermediate cylinders, cubic feet	9.74
Capacity of the receiver between the intermediate and the large cylinders, cubic feet	53.80
Diameter of the air pump (one, single acting and lifting), inches	26
Stroke of the air pump piston, feet	2
Area of the air pump piston, square inches	530.930
Space displacement of the air pump piston per stroke, cubic feet	7.374
Number of iron tubes in surface condenser (each engine)	510
Outside diameter of surface condenser tubes, inches	1.25
Length of surface condenser tubes, feet	7.57
Aggregate surface of outside of condenser tubes, square feet	1,053
Number of pumps, vertical and single acting	3
Diameter of the plungers of the pumps, inches	30
Stroke of the plungers of the pumps, feet	4
Aggregate area of the plungers of the three pumps, square inches	2,120.58
Aggregate space displacement of the plungers of the three pumps, cubic feet	58.905
Ratio of the aggregate area of the plungers of the three pumps, to the area of the piston of the large cylinder, allowing for the single action of the pumps and the double action of the cylinder	2.397

MANNER OF MAKING THE EXPERIMENTS.

Before commencing the experiments the observers devoted considerable time to securing accuracy for the indicator diagrams, to producing proper circulation of the steam through the jackets of the cylinders, to calibrating the feed water tanks and the tanks which received the water of steam liquefaction drained from the jackets of the several cylinders and from the steam pipe, and to testing the tightness of the steam pistons, the steam valves, and the joints of the jackets.

Each jacket was fitted with an air cock at its upper extremity, to be operated by hand for the discharge of any air that might accidentally be present. The jackets of each cylinder had only one steam supply pipe, one drain pipe, one steam trap, one reducing valve, and one pressure gauge in common. During the experiments the reducing valve was not used, and the full boiler pressure was maintained in all the jackets of all the cylinders as nearly as the proportions of the supply pipe would admit, the strength of the cylinders and jackets being sufficient for that pressure. The water of steam liquefaction from the jackets of each cylinder was passed through a steam trap, collected and measured for each cylinder as an aggregate.

The steam pipe was of considerable length, and a portion of it was bare. Whatever water of steam liquefaction was in this pipe, was drained off near the engines into a well pipe and steam trap, from which it was delivered into a tank and measured.

The pressures in the cylinders were measured from indicator diagrams taken by Richard indicators. For the small and the intermediate cylinders, the indicators were coupled to three-way taps placed half way up the cylinders and connected to the cylinder ends by well clothed pipes. This arrangement was only adopted after it had been ascertained by trial to produce absolutely identical diagrams with those taken when the taps were screwed directly into the ends of the cylinders. In the case of the large cylinders, the indicators were coupled directly to the ends of the cylinders, as trial showed the diagrams to be lessened in area when the indicators were employed in connection with pipes. The springs for the indicators on the small cylinders

were set to the pressure of 60 pounds per square inch, those for the indicators on the intermediate cylinders were set to the pressure of 30 pounds per square inch, and those for the indicators on the large cylinders were set to the pressure of 10 pounds per square inch. The springs of the indicators were tested after the completion of the experiments. At the temperature of 70 degrees Fahrenheit they were practically correct, and at the temperature of 190 degrees they were about one and a half per centum too weak. Considering the uncertainty which exists as to the temperature to which indicator springs are exposed, the diagrams from the small and intermediate cylinders were reduced by lessening the height of each ordinate two per centum. No correction was made for the diagrams from the large cylinder.

The head against which the pumps delivered was ascertained by means of a Richard indicator placed on the delivery main, and to the pressure thus shown was added the pressure due to the height of the indicator above the water level in the pump well. The pumps delivered their water directly into the main, and not into a reservoir.

The dimensions of the engines were given by their makers, Messrs. Yates and Thom of Blackburn; the capacities of the clearance and steam passages at one end of each cylinder, and the surfaces of the same, as well as the jacketed surfaces, were calculated from the drawings of the cylinders.

The boiler pressure and the pressures in the jackets were taken by gauges which were not tested, but which were believed to be very nearly accurate. The vacuum gauges on the condensers were mercurial columns. The number of revolutions made by each engine was taken by a counter attached to it.

The feed water was measured in two permanent concrete tanks, each 6 feet 6 inches long, 1 foot 6½ inches wide, and 4 feet 5 inches deep. They were calibrated with a narrow necked tin can that held exactly 100 pounds of water at the temperature of 65 degrees Fahrenheit. The weight measured into each tank was 2,000 pounds, occupying a depth of 3 feet 2⅝ inches. These tanks were filled alternately, the time when each was emptied being noted. From these measuring tanks the water was discharged into a larger

tank from which the feed pump drew. This tank, which may be called the feed tank, was 12 feet long and 3 feet 10½ inches wide. The water level in it was ascertained by means of a float carrying a scale graduated in tenths of an inch. The float rose and fell in a cast-iron pipe placed vertically in the tank with the upper end above the water level, so that the float might not be disturbed by oscillations of the water. From the feed tank the water was pumped into the two boilers that supplied the engines with steam.

The experiments were commenced and ended with the water level and the steam pressure in the boilers exactly the same, and were limited to the six hours of the day when the consumption of water from the main was most uniform. The boilers were abundantly large for the work, and furnished dry steam. Indicator diagrams, and observations, were taken by numerous and competent assistants of all the quantities given in the following table. No regard was paid to the performance of the boilers, or to the economy of the coal consumed. With the large water heating surface in the boilers, and the slow rate of evaporation per square foot of that surface, the water level in the boiler gauge glasses was remarkably steady, admitting of very exact measurement.

The two boilers used were of the Lancashire type, each was 30 feet long and 7½ feet diameter, set in brick masonry and working in connection with one economizer of 192 cast-iron tubes. The aggregate water level surface was 396 square feet. Every possible water leakage, however insignificant, was detected, measured and deducted.

The experiments were made simultaneously on both engines, both being supplied with steam from the same two boilers. These engines are respectively designated the "North Engine" and the "Middle Engine."

The two experiments made—one on the 1st of March and the other on the 9th of March—were duplicates, and their results show the limits within which the experimental errors were confined, these limits were less than two per centum for the extremes, and less than one per centum for the means, an exceedingly close agreement, proving the great accuracy with which both experiments were conducted.

EXPLANATION OF THE FOLLOWING TABLE CONTAINING THE DATA AND RESULTS OF THE TWO EXPERIMENTS MADE BY MR. MICHAEL LONGRIDGE SIMULTANEOUSLY ON THE TWO TRIPLE EXPANSION, DIRECT ACTING, STEAM JACKETED, SURFACE CONDENSING, VERTICAL PUMPING ENGINES AT THE EAST LONDON WATER WORKS, LEA BRIDGE, ENGLAND.

In the following table the quantities, experimental and deductive, are arranged in groups and on numbered lines for facility of reference. The descriptions of the quantities on the lines are so full that but little additional explanation is necessary.

Total Quantities.—Lines 1 to 9, both inclusive, contain the total quantities experimentally ascertained in the manner previously described. The quantity on line 8 is the sum of the quantities on lines 5, 6 and 7.

The quantity on line 10 was obtained by subtracting from the quantity on line 4, the sum of the quantities on lines 8 and 9. This quantity (line 10) is the total weight of steam in pounds that entered the small cylinders of both engines simultaneously during each experiment, and is exclusive of the weights of steam liquefied in all the jackets of all the cylinders, and of the weight of steam liquefied in the steam pipe.

The quantity on line 11 is the quotient of the quantity on line 5 divided by the quantity on line 4.

The quantity on line 12 is the quotient of the quantity on line 6 divided by the quantity on line 4.

The quantity on line 13 is the quotient of the quantity on line 7 divided by the quantity on line 4.

The quantity on line 14 is the sum of the quantities on lines 11, 12 and 13.

The quantity on line 15 is the quotient of the quantity on line 9 divided by the quantity on line 4.

The quantities on lines 5 to 15, both inclusive, with the exception of the quantity on line 10, concern the waters of steam liquefaction drawn from the steam jackets of the cylinders and from the steam pipe; these quantities relate to these waters as weights of water only; they do *not* represent quantities of heat proportionally to the quantity of heat in the feed water vaporised in the boilers (line 4), because the heat taken out of them is

only the latent heat of the steam of the respective pressures under which the liquefaction was effected, whereas, the heat in the water vaporized in the boilers (line 4) contains all the heat above the temperature of the feed water (line 65). The quantities on lines 10 and 16, however, represent both as water weights and as heat units relatively to the quantity on line 4, the proportion of the total weight of steam entering the small cylinders, to the total weight of steam evaporated in the boilers. The quantity on line 16 is the quotient of the quantity on line 10 divided by the quantity on line 4.

The quantities on lines 17, 18 and 19, were obtained by direct measurement, but how performed is not described in Mr. Longridge's report. They are important quantities, being the total number of pounds of water of liquefaction drawn respectively from the jackets of the small, the intermediate, and the large cylinders, due to their external radiation of heat. The trials by which they were obtained were not, of course, made simultaneously with the experiments on the engines doing work, but both before and afterwards with the engines at rest, the same pressure being maintained in the jackets that was there during the experiments with the engines at work. The trials were made on both engines separately at different times. The trial made on "North Engine" commenced at 5 hours 40 minutes P. M. on the 24th of February, and ended at 10 A. M. on the 25th, lasting sixteen and one-third consecutive hours. The trial made with the "Middle Engine" commenced at 10 hours 48 minutes A. M. on the 10th of March and ended at 2 hours 13 minutes, lasting consecutively three hours and twenty-five minutes. The quantities on lines 17, 18 and 19, are the averages of the hourly means given by both trials, multiplied by the number of hours of the experiments made with the engine at work, so that these quantities are proper for the two engines when working simultaneously. Strictly speaking, the trials should have been made with the cylinders filled with steam of the average pressure which was in them while the engines were working, as well as the jackets should have been filled with steam of the pressure in them while the engines were working. Also, the temperature of the air

surrounding the engines should have been the same in both cases, but the report is silent as to whether such were the facts or not.

Line 20 contains the sum of the quantities on lines 17, 18 and 19.

The weights of water on lines 17, 18, 19 and 20, do not represent quantities of heat relatively to the heat imparted to the weight of water vaporized in the boilers (line 4), because the waters of liquefaction from the jackets had the temperature due to the steam pressure in the jackets, while the water vaporized in the boilers received heat above the temperature of the feed water (line 65), a much less temperature than that of the jacket water. The steam in the jackets parted with only its latent heat, while the heat imparted to the water in the boilers contained, in addition to the latent heat of the boiler steam, the heat required to raise the water from the temperature of the feed (line 65) to that of the boiler steam (line 51).

The quantities on line 21 are the quotients of the division of the quantities on line 20 by the quantities on line 4.

Engines.—The quantity on line 22 is the quotient of the quantity on line 4 divided by the number of hours on line 1. And the quantity on line 23 is the quotient of the quantity on line 10 divided by the same number of hours.

The quantity on line 24 is the number of Fahrenheit units of heat that would have been imparted to the feed water in the boiler (line 4), had its temperature been the temperature on line 65 instead of the temperature on line 61. The boiler being fed during the experiments from the main, the feed water (line 4), had the experimental temperature on line 61, the waters of steam liquefaction from all the jackets of all the cylinders (line 8), and from the steam pipe (line 9), being wasted, that is, not returned to the boilers, but when the engines were in regular work, these waters were fed to the boilers as part of the feed water, and their greater temperature than the condenser temperature (line 63) proportionally increased the temperature of the entire feed, so that in the case of the experiments had these waters been used as part of the feed, the temperature of the feed water would have been the tempera-

ture on line 65, which temperature is exclusive of an allowance of 3.9 degrees supposed lost by external radiation from the feed pipe between the engines and boilers. In determining the number of Fahrenheit units of heat imparted to the feed water (line 24), the calculation has been made not simply for differences of temperature of the waters, but for differences of units of heat normal to the different temperatures.

Lines 25 and 26 are the quotients respectively of the quantities on lines 2 and 3 by the number of minutes on line 1. The quantity on line 27 is the average of the quantities on lines 25 and 26. The variation between these quantities is too slight to be productive of any sensible effect.

The quantity on line 28 is the product of the quantity on line 27 multiplied by 8 feet, or twice the stroke of the steam pistons.

The quantities on lines 29 to 39, both inclusive, were by direct observation of gauges.

The pressure in the steam jackets of the small cylinders (line 32), was 4 pounds per square inch less than the boiler pressure (line 30), in both experiments.

The pressure in the steam jackets of the intermediate cylinders (line 34), was 5 pounds per square inch less than the boiler pressure for the experiment of March 1st, and 6 pounds per square inch less for the experiment of March 9th.

The pressure in the steam jackets of the large cylinders (line 36), was 7 pounds per square inch less than the boiler pressure in both experiments.

There will be observed that the difference between the pressure in the boilers and in the jackets differs for the different cylinders, increasing as the capacities of the cylinders increase. For the small cylinders the difference is 4 pounds per square inch; for the intermediate cylinders it is (mean of both experiments) 5½ pounds; and for the large cylinders it is 7 pounds. Supposing the steam for the jackets of all the cylinders to be taken from the same place—either boiler or steam pipe—by means of small pipes of the same diameter, which was probably the case, the increasing difference of pressure for the different cylinders must have been due to the insufficient cross area of these pipes. As

the cylinders enlarged, not only was a greater volume of steam required for the jackets, but the liquefaction in the jackets became greater per square foot of jacket surface because of the greater difference between the jacket pressure and the mean steam pressure in the cylinders due to the expansion of the steam in the latter. The volume of steam in the case of each cylinder that was passed through its respective jacket pipe, was the volume due to the weight of the water of liquefaction drawn from its jackets and to the pressure in those jackets. Had the cross area of the pipes been given, an experimental determination of value could have been made in this respect. A report should contain every detail and dimension of the apparatus. From the foregoing data as regards the steam jacket pressure, will be seen the necessity of always having a pressure gauge upon every jacket, as the usual inference in the absence of a gauge that the boiler pressure is present in the jacket, is wholly fallacious. The jacket pressure may be, and generally is, much lower than the boiler pressure. The writer knows of cases wherein the jacket pressure was scarcely one-half the boiler pressure, and yet the cylinders of these jackets were supposed to be steam jacketed with boiler pressure steam. The proof tests were made in consequence of the writer's doubts, based on the economic indications, that there was any reasonably close approach to the boiler pressure in the jackets.

The initial pressure in the small cylinders was 6.3 pounds per square inch less than the boiler pressure; the length of the steam pipe and its diameter are not given. If the steam for the jackets was taken from the steam pipe near the small cylinders, a part of the discrepancy between the boiler pressure and the jacket pressure would be accounted for, but not the increasing discrepancy due to the increasing capacities of the cylinders and their diminishing steam pressure. But no matter how large the cross area of the pipes supplying the jackets with steam, or how short their lengths, and supposing them to bring the steam direct from the boilers, they could not deliver the boiler pressure into the jackets; there would always be a less pressure in the latter than in the boiler, how much less depending on whatever modifying cir-

cumstances existed. Hence, under no conditions can the inference be accepted that the pressure in the jackets is equal to the pressure in the boilers, and any calculations based on it must always be false in some degree, and perhaps largely false. There should, in all experiments involving jacketing, be a pressure gauge upon the jacket and its indications recorded.

When steam is generated in a boiler in free communication with the cylinder of a working steam engine, it begins to expand at the moment it is generated, and continues to expand until the cut off valve closes in the small cylinder, if the engine be a multiple cylinder one, the reduction of pressure being continuous from the heating surface in the boiler up to the instant that the cylinder communication with the boiler is interrupted. As regards the jacket, the reduction of the pressure is continuous from the heating surface of the boiler to the liquefaction of the steam in the jacket. This condition of continuously decreasing pressure is essential for the continuous flow of the steam. With equality of pressure there could be no current anywhere, such equality being predicable for an elastic fluid only when that fluid is at rest.

The temperature of the water of liquefaction in the jackets is necessarily less than the temperature of the steam in them. This water is deposited on all the inner surfaces of the jacket, on those opposite the cylinder as well as those of the cylinder, and from these surfaces it trickles to the bottom of the jacket whence it is drawn off by a pump or by a trap. This water must, of course, have the temperature of the surfaces on which it was deposited, and not the higher temperature of the steam from which it was precipitated; the only cause of its precipitation was the lower temperature of these surfaces; and that lower temperature was caused by the facts that they radiated heat externally to the surrounding air and imparted it to the steam within the cylinder. How much the temperature of the water is below the temperature of the steam can be ascertained only by measurement, and the water temperature should always be thermometrically determined in the case of exact experiment. The temperature of the feed water as it enters the boilers should also be thus ascertained

instead of being assumed to be the same as the temperature of the hot-well, whereas it is always less, and, in many cases, much less.

The quantities on lines 40 to 48, both inclusive, are taken directly from the indicator diagrams, and are the means given by all these diagrams.

Line 49 gives the number of times the steam was expanded in the small cylinders, calculated by first adding the length of the stroke of the piston, taken as unity, to the fraction of the stroke equivalent to the waste spaces at one end of the cylinder, and then dividing this sum by the fraction of the stroke of the piston completed when the steam was cut off (line 40) plus the fraction of the stroke equivalent to the waste spaces at one end of the cylinder.

Line 50 gives the number of times the steam was expanded in the engines, calculated by multiplying the quantity on line 49 by the ratio which the space displacement of the piston of the large cylinder per stroke plus the waste spaces at one end of that cylinder, is to the space displacement of the piston of the small cylinder per stroke plus the waste spaces at one end of the small cylinder. The intermediate cylinders need not be considered.

The quantities on lines 49 and 50 cannot be exact, owing to the complicated liquefactions of the steam and to the irregular revaporizations of the waters of these liquefactions in the cylinders, and to the superheatings given by the excess of the temperature of the steam in the jackets. What is exactly shown is the ratio of the spaces previous and subsequent to the closing of the cut-off valves on the small cylinders. The real expansions undergone by the steam must be greater in a marked degree than those on lines 49 and 50.

Temperatures.—The temperatures are all given in Fahrenheit degrees. Lines 51, 52, 53 and 54 give, respectively, the temperatures normal to the steam pressures in the boilers and in the steam jackets of the different cylinders.

Lines 55, 56 and 57 give the mean temperatures of the steam in the small, the intermediate, and the large cylinders. These temperatures are normal to the mean of the entire cycle of pres-

sures in each cylinder from the commencement of the steam stroke of the piston to the end of the exhaust stroke, these pressures being taken from the indicator diagrams.

The quantity on line 58 is the difference between the quantities on lines 52 and 55.

The quantity on line 59 is the difference between the quantities on lines 53 and 56.

The quantity on line 60 is the difference between the quantities on lines 54 and 57.

The quantities on lines 61, 62 and 63, were given directly by the thermometer.

The quantity on line 64 is the temperature the feed water (line 4) would have had, had the waters of liquefaction from all the jackets of all the cylinders, and from the steam pipe (lines 5, 6, 7 and 9) been used for part of the feed, the remaining part consisting of water taken from the hot well at the temperature of 72 degrees (line 63). In this calculation the temperatures of the waters of liquefaction from the jackets have been taken as normal to the respective jacket pressures, and the temperature of the water of liquefaction from the steam pipe has been taken as normal to the boiler pressure. Consequently, the quantities on line 64 are somewhat in excess of the truth, but the data allows no nearer an approach.

Line 65 contains the *assumed* temperature at which the feed water would have entered the boilers under normal conditions. This quantity is the quantity on line 64 less 3.9 Fahrenheit degrees, allowed for loss of temperature by radiation from all the feed pipe surfaces, etc., between the condensers and the boilers. From the temperature on line 64, and the boiler pressure on line 30, the heat imparted to the feed water in the boilers has been calculated, lines 24, 66 and 67.

Heat.—The quantity on line 66 is the product of the multiplication of the quantities on lines 22 and 24.

The quantity on line 67 is the product of the multiplication of the quantities on lines 23 and 24.

The quantity on line 68 is the product of the multiplication of the quantity on line 17 by the latent heat of steam of the pres-

sure in the jackets of the small cylinders (line 32), divided by the number of hours on line 1.

The quantity on line 69 is the remainder of the quantity on line 80 after subtraction of the quantity on line 68.

The quantity on line 70 is the quotient of the quantity on line 69 divided by the number of square feet (59) of jacketed surface in the small cylinders.

The quantity on line 71 is the product of the multiplication of the quantity on line 18 by the latent heat of steam of the pressure in the jackets of the intermediate cylinders (line 34) divided by the number of hours on line 1.

The quantity on line 72 is the remainder of the quantity on line 81 after subtraction of the quantity on line 71.

The quantity on line 73 is the quotient of the quantity on line 72 divided by the number of square feet (101.8) of jacketed surface in the intermediate cylinders.

The quantity on line 74 is the product of the multiplication of the quantity on line 19 by the latent heat of steam of the pressure in the jackets of the large cylinders (line 36) divided by the number of hours on line 1.

The quantity on line 75 is the remainder of the quantity on line 82 after subtraction of the quantity on line 74.

The quantity on line 76 is the quotient of the quantity on line 75 divided by the number of square feet (176) of jacketed surface in the large cylinders.

The quantity on line 77 is the quotient of the quantity on line 70 divided by the quantity on line 58.

The quantity on line 78 is the quotient of the quantity on line 73 divided by the quantity on line 59.

The quantity on line 79 is the quotient of the quantity on line 76 divided by the quantity on line 60.

The quantities on line 77, line 78 and line 79, are the number of Fahrenheit units of heat passed per hour respectively from the steam in the jackets of the small, the intermediate, and the large cylinders into the steam within those cylinders per square foot of their jacketed surface and per degree Fahrenheit of difference between the temperature of the steam in the jackets and within

the respective cylinders. Now to test the quantities on the above three lines, there are required the additional data of the densities of the steam of the different pressures, and the different specific heats of this steam; these densities and specific heats entering inversely into the problem. Other things being equal, the less the density of the recipient the slower the flow of the heat into it, and the less the specific heat of the recipient the slower the flow of heat into it. The transfer of heat per unit of time per unit of surface from the steam in the jacket to the less pressure steam within the cylinder, will be in the ratio of the difference of temperature of the two steams multiplied by the difference of their densities, and the product multiplied by the difference of their specific heats. This law of the transfer of heat from steam of one pressure to steam of a less pressure, applies, of course, to dry steam only for both the delivering and the receiving steam, and the effect is to superheat the latter. In the case of an unjacketed multiple cylinder engine, such steam is found only in the small cylinder previous to the closing of the cut-off valve. Subsequent to that closing the steam is always wet, owing to the liquefaction of a portion of it to furnish the heat transmuted into the power developed by the expanding steam, a liquefaction which takes place molecularly, that is upon a portion of each molecule, so that the water of the liquefaction is uniformly disseminated among the steam and remains suspended in it producing the appearance of fog or mist. Steam doing work expansively is the only kind of wet steam; the only kind in which the appearance of fog or mist is produced. Steam doing work without expansion remains perfectly transparent and dry.

The liquefaction of steam in a cylinder due to the radiation of heat, occurs solely at the radiating metallic surfaces, and the water of such liquefaction is deposited on those surfaces only, unintermingled with steam. The water is a mass by itself, and the steam is another mass by itself. In the case of "foaming" or "priming," the water comes over from the boiler into the cylinder in masses accompanying the steam but not intermingled with it. The steam and the water occupy distinct spaces, the water flows along the bottom of the steam pipe with the steam

flowing along above it, the two forming separate masses without uniform intermixture. The same remarks apply to the liquefaction of steam in cylinders, produced by the interaction of the heat of the steam with the metal of the cylinder. The deposition of the water of this liquefaction is entirely and in mass upon the metallic surfaces.

If the small cylinder be jacketed with steam of sufficiently high temperature to more than prevent liquefaction previous to the closing of the cut-off valve, then the additional heat transferred from the jacket to the steam within the cylinder will superheat that steam, in which case the specific heat will be practically constant at 0.48. The steam in the remainder of the stroke of the piston of the small cylinder, would necessarily be superheated also, and would not contain any water of the molecular liquefaction due to the expansive working. Steam cannot be superheated and contain this water at the same time. Superheated steam is not a vapor but a gas, and the heat transmuted into the work done by the expansion of such gas produces a corresponding lowering of the temperature of the superheated steam, but no precipitation of water. When the precipitating point has been reached, the steam is a vapor again; on either side of this point, it is subject to different laws. If the heat received by the steam within the small cylinder from the steam in the jacket previous to the closing of the cut-off valve be not sufficient to prevent surface liquefaction, the water of such liquefaction will be revaporized during the remaining portion of the stroke of the piston and during the following exhaust stroke of the piston, the heat being supplied partly from the contained heat in the water rendered available by the lessening of the pressure upon it, and partly from the heat in the jacket steam.

Thus, in the actual steam engine, the heat transferred from the steam in the jacket, is transferred to both water and steam in the cylinder, vaporizing the former and superheating the latter. The ratio of the weight of water to the weight of steam is, of course, a very variable one, depending on many circumstances. No superheating, however, can be done until all the water present in the cylinder, however derived, is vaporized. Under such complex and

varying circumstances, unknown and unknowable, no calculation can be made as regards the quantitative distribution of the heat taken from the steam in the jacket to the steam within the cylinder.

An important result of these experiments is the showing of how large a portion of the heat taken from the jacket steam is due to external radiation, which, of course, must be subtracted from the total loss of heat by the jacket steam to give the heat imparted to the steam and water within the cylinder. A most erroneous idea will be formed if all the heat lost by the jacket steam be supposed transferred to the water and steam in the cylinder. The economy produced by jacketing is effected by much less heat than is commonly imagined.

The quantity on line 80 is the product of the multiplication of the quantity on line 5 by the latent heat due to the steam pressure on line 32 divided by the number of hours on line 1.

The quantity on line 81 is the product of the multiplication of the quantity on line 6 by the latent heat due to the steam pressure on line 34 divided by the number of hours on line 1.

The quantity on line 82 is the product of the multiplication of the quantity on line 7 by the latent heat due to the steam pressure on line 36 divided by the number of hours on line 1.

The quantity on line 83 is the sum of the quantities on line 80, line 81 and line 82.

The quantity on line 84 is the product of the multiplication of the quantity on line 22 by the quantity on line 24.

The quantity on line 85 is the fraction which the quantity on line 83 is of the quantity on line 84.

The quantity on line 86 is the sum of the quantities on line 68, line 71 and line 74.

The quantity on line 87 is the remainder of the quantity on line 83 after subtraction of the quantity on line 86.

The quantity on line 88 is the fraction which the quantity on line 86 is of the quantity on line 84.

The quantity on line 89 is the fraction which the quantity on line 87 is of the quantity on line 84.

Steam Pressures in the Small Cylinders.—The quantities on

lines 90 to 100, both inclusive, are taken by measurement from the indicator diagrams.

Steam Pressures in the Intermediate Cylinders.—The quantities on lines 101 to 112, both inclusive, are taken by measurement from the indicator diagrams.

Steam Pressures in the Large Cylinders.—The quantities on lines 113 to 124, both inclusive, are taken by measurement from the indicator diagrams.

Equivalent Steam Pressures for the Large Cylinders.—The quantities on lines 125 to 129, both inclusive, show what would be the indicated, net, total and back pressures on the pistons of the large cylinders, were these pressures on the pistons of the small and intermediate cylinders reduced in the ratio of the net area of the pistons of the large cylinders to the net areas respectively of the pistons of the small and intermediate cylinders, and the quotients added to the respective experimental pressures on the pistons of the large cylinders.

In all multiple cylinder engines, the dimensions of the large cylinder are the same as those of the single cylinder of a single cylinder engine using the steam with the same regimen as the multiple cylinder engine. For the purpose of comparison, moreover, the various pressures on the pistons of the multiple cylinder engine, must be reduced to a single expression.

The total pressure on line 127 supposes no cushioning of the back pressure, consequently, the quantity on line 128, being the difference of the quantities on lines 125 and 127, is the back pressure supposing no cushioning. With the experimental cushioning in the large cylinder (beginning at 0.75 of the return or exhaust stroke of the piston of that cylinder) the back pressure for this 0.75 of the stroke would be the quotient of the quantity on line 128 divided by 0.75, which quotient is the quantity on line 129. If, now, the power due to the indicated pressure on line 125 be calculated, using the entire area of the piston of the large cylinder; and if the power due to the back pressure on line 129 be calculated, using 0.75 of the area of the piston of the large cylinder, the sum of these two powers will equal the aggregate total power, line 150, developed by the engine.

A comparison of the experimental equivalent total pressure (line 127) on the pistons of the large cylinders, with the corresponding calculated total pressure which would be obtained on the supposition that the steam expanded according to the law of Mariotte, will be interesting and instructive, as this calculated pressure would be the mean total experimental pressure obtained in the single cylinder of a single cylinder engine using the steam with the same regimen as in the three cylinders of the triple expansion engine.

The mean pressure of steam during its expansion in the single cylinder of a single cylinder engine, as shown by the results of a vast number of indicator diagrams taken with steam of widely varying regimen in cylinders of widely varying dimensions, both jacketed and unjacketed, agrees almost exactly with the mean pressure due to the expansion, as calculated by the Mariotte law. The coincidence is, of course, purely accidental, arising from the numerous and complex conditions acting upon the steam pressure in a working cylinder, some of which increase while others decrease the pressure, the final result, as a balance of the opposing conditions, causing the pressure to be what it would be if calculated according to the Mariotte law. This coincidence is of great practical value, allowing many important comparisons to be conveniently made as regards the performances of engines, which would otherwise be difficult and uncertain.

If the steam used in the experimental triple expansion engine, were used with exactly the same regimen in the single cylinder of a single cylinder engine, having the dimensions of the large cylinder of the experimental engine, the following would be the data for such a single cylinder :

Fraction completed of the stroke of the piston when the steam was cut off,	0.0509
Fraction completed of the exhaust stroke of the piston when the cushioning of the back pressure began	0.75
Fraction of stroke of piston displacing the same space as the aggregate spaces in the clearance and steam passage at one end of the cylinder, and in the cylinder up to the point of cutting off	0.056
Number of times the steam was expanded, line 50	18.385
Mean steam pressure in pounds per square inch above zero on the piston previous to the point of cutting off, line 99	128.129

Steam pressure in pounds per square inch above zero on the piston at the point of cutting off, line 91	119.500
Mean pressure of the expanding steam alone, in pounds per square inch above zero, subsequent to the point of cutting off, calculated according to the Mariotte law	20.732
Mean steam pressure in pounds per square inch above zero on the piston during its entire stroke, according to the Mariotte law for the expansion portion of the stroke	26.197

The experimental mean pressure of the expanding steam in the cylinders of the triple expansion engine, was 17.903 pounds per square inch above zero, being (20.732—17.903=) 2.829 pounds per square inch less than the mean pressure of the expanding steam calculated by the Mariotte law, or $(\frac{2.829}{20.732}=)$ 0.1365 of the calculated expansion pressure alone.

The experimental mean pressure of the steam during the entire stroke of the pistons in the cylinders of the triple expansion engine, was 23.5114 pounds per square inch above zero, line 127, being (26.1969—23.5114=) 2.6854 pounds per square inch less than the mean pressure during the entire stroke of the pistons calculated for the expansion portion of the stroke according to the Mariotte law, or $(\frac{2.6854}{26.1969}=)$ 0.1025 of the calculated mean pressure for the entire piston stroke.

These are very serious losses and are referred to the total horses-power developed by the engines, line 150, for the latter fraction, and to the horses-power developed by the expanding steam alone after the closing of the cut-off valve on the small cylinder, line 151, for the former fraction. When they are referred to the indicated and to the net horses-power developed by the engine, and to the horses-power doing the net pumping, they become still more serious; for instance:

The mean experimental equivalent indicated pressure on the pistons is, for the experiment of March 9th, 19.699 pounds per square inch, line 125. The mean calculated equivalent indicated pressure, with the same back pressure, line 128, of 3.812 pounds per square inch, is (26.197—3.812=) 22.385 pounds per square inch, a difference of 2.685 pounds per square inch, as before, but now amounting to $(\frac{2.685}{22.385}=)$ 0.1199 of the indicated horses-power, line 148.

The mean equivalent experimental net pressure on the pistons is, for the experiment of March 9th, 18.622 pounds per square inch, line 126. The mean calculated equivalent net pressure, allowing the equivalent pressure on the pistons of the large cylinders of 1.077 pounds per square inch for working the engines *per se*, or unloaded, is (22.385 — 1.077 =) 21.307 pounds per square inch, a difference of (21.307—18.622=) 2.685 pounds per square inch as before, now amounting to $(\frac{2.685}{21.307}=)$ 0.126 of the net horses-power, line 149.

The mean equivalent experimental pressure in pounds per square inch on the pistons of the large cylinders, of the pressure in the main just outside of the delivery valves of the pumps, supposing the pumps to fill to 0.98 of their capacity, is, for the experiment of March 9th, 17.160, line 131. The mean equivalent calculated pressure, will therefore be (17.160+2.685=) 19.845 pounds per square inch, and the difference of the 2.685 pounds per square inch will amount to $(\frac{2.685}{19.845}=)$ 0.1353 of the net horses-power expended in the pumping, line 132.

As the object of a pumping engine is to pump, the economy of the pumping is the commercial value of the performance, consequently, the economy of the experimental engine was lessened 13½ per centum by the less mean pressure of the steam during the stroke of the pistons of the triple expansion engine, than the corresponding mean pressure would have been in the case of a single cylinder engine with a single cylinder of the dimensions of the large cylinder of the triple expansion engine, the steam being used with the same regimen in both cases.

This fraction represents a very important loss for the triple expansion engine, consequently, the tracing of it to its cause is an equally important investigation. The loss results wholly from the too small capacity of the receivers between the small and intermediate cylinders, and between the intermediate and large cylinders. The too little capacity of the receivers relatively to the capacity of the cylinders in connection with them, causes the steam passing through the receivers to undergo expansion in them, so that the steam pressure in the cylinder drawing steam from a receiver is less than it should be, and less than it would be

had the steam not thus expanded. Also the back pressure in the cylinder delivering steam into a receiver is greater than it should be, and greater than it would be if the receiver was of larger capacity. A complete experiment on a multiple cylinder engine includes indicator diagrams taken from its receivers, as these diagrams are the only means of showing the degree of expansion undergone by the steam in passing through the receivers. A pressure gauge (wanting in the present experiments) on the receivers shows only the mean pressure in them, not the initial and terminal pressures, which are the pressures needed. Such diagrams are rarely ever, perhaps never taken, but if taken they would afford conclusive evidence of the expansion in question, and, if it exist, then the work that might have been obtained from it is as surely lost. The proper capacity for any receiver is, strictly speaking, infinite, as in that case alone there would be no expansion of the steam in passing through it, and therefore no loss from such expansion. The larger the capacity of the receiver the less will be the loss due to the expansion of the steam passing through it, but, on the other hand, the larger the capacity of the receiver the greater will be the extent of its surface exposed to external radiation of heat operating liquefaction of the steam within, and thus causing an economic loss of another kind. There is, therefore, a capacity of receiver, differing for different cases and conditions, in which the sum of the losses by the expansion of the steam within it, and by its external radiation of heat is a minimum. In general practice, owing to convenience of construction, money cost, and ignorance, the capacities of receivers are much below the limit required to obtain this minimum. The greater the number of cylinders employed to form an engine, the greater will be the losses of both kinds due to the use of receivers. These losses, of course, do not exist for the single cylinder engine.

The loss of useful effect by the expansion of the steam in the receiver, is additional to the loss of useful effect by the difference between the back pressure, exclusive of the cushioning, in the cylinder delivering into the receiver, and the steam pressure up to the point of cutting off in the cylinder taking steam from the receiver; such difference constitutes additional back pres-

sure to the experimental back pressure, exclusive of cushioning, in the large cylinder. There would be a loss from this additional back pressure, even were the receiver infinite in capacity, because in the practical multiple cylinder engine the steam can get out of the one cylinder into the succeeding cylinder only by a difference of pressure measured by the resistance of the steam ports and passages of the two cylinders. If these steam ports and passages were made so large in area and so direct as to offer no sensible obstruction to the flow of the steam, the back pressure exclusive of cushioning in the delivering cylinder would be the same as the steam pressure up to the point of cutting off in the receiving cylinder.

The effective or equivalent back pressure in a multiple cylinder engine, the condenser pressure remaining constant, can be made very small or very large according to the area of the steam ports; and the cross area, length and sinuosity of the steam passages connecting the cylinders; and to the reciprocating speed of the piston; and to the capacity of the receiver. The expansion of the steam in the receivers can likewise be made much or little according to the same conditions. Hence, the economic performance of multiple cylinder engines is greatly affected by the proportions of the parts just enumerated, so that variations of from 10 to 20 per centum are easily possible between different engines of the same type working with substantially the same regimen of steam, but having these parts differently proportioned, and more especially in cases where the steam having an excessive measure of expansion, has necessarily a small mean indicated equivalent pressure when referred to the large cylinder, in which cases the losses of effective pressure by the additional back pressure due to the transfer of the steam from one cylinder to the succeeding cylinder, and by the expansion of the steam in the receiver, become a large percentage of the mean pressure on the piston during its stroke. The economic performances of single cylinder engines, not being influenced by the variable effects due to the employment of receivers, are much more uniform when the steam is used with the same regimen.

Pumps.—A portion of the quantity on line 130, ascertained by means of a pressure gauge in the main just outside of the delivery valves of the pumps, is the pressure equilibrating the mean pressure in the main during the experiments, the remaining portion being the resistance of the statical weight of a column of water having a vertical height equal to the vertical distance apart of two horizontal planes, one passing through the center of the orifice for the gauge, and the other coinciding with the surface of the water in the pump well. The aggregate pressure, line 130, is less than the pressure against which the pistons of the pumps act by the pressures due to the resistance of the surfaces moved over by the water between the gauge and the lower edge of the suction main, to the resistance caused by the sinuosities of those surfaces, and to the resistance due to the opening of the receiving and delivering valves of the pumps, and to the forcing of the water through these valves. The quantity on line 130 may be considered the net resistance against which the pump pistons pump, that is, the resistance due to the mere weight of the water alone, and exclusive of the resistances due to the pumping mechanism alone. Strictly speaking, there should be included in the latter, additionally, the frictions of the pump pistons and the pump stuffing boxes. In the present case, however, these frictions are included in the pressure required to work the engines *per se* or unloaded.

The pressure on line 131 is the pressure on line 130 divided by the ratio of the net areas of the pistons of the large cylinders to the net areas of the pistons of the pumps and the quotient multiplied by 0.98, this being the assumed fraction of the capacity of the pumps to which they will fill with water, the remaining 0.02 being what is technically known as the "slip" of the pump.

The horses power on line 132 is calculated for the pressure on line 130 multiplied by 0.98.

The quantity on line 133 is the difference of the quantities on lines 126 and 131.

The quantity on line 134 is the fraction which the quantity on line 132 is of the quantity on line 148.

The quantity on line 135 is the difference between the fraction on line 134 and unity.

Horses-Power.—Line 136 contains the number of indicated horses-power developed in the small cylinders. They are calculated for the entire net area of the pistons of these cylinders, for the piston speed on line 28, and for the indicated pressure on line 96.

Line 137 contains the number of net horses-power developed in the small cylinders. They are calculated for the entire net area of the pistons of these cylinders, for the piston speed on line 28, and for the net pressure on line 97. This power is exclusive of the power required to work the engines *per se*, or unloaded, but it includes the power due to the pressure caused upon the working surfaces of the fly-wheel mechanism by whatever portion of the load is passed through that mechanism.

Line 138 contains the number of total horses-power developed in the small cylinders exclusive of the power required to overcome the cushioning. They are calculated for the entire net area of the pistons of these cylinders, for the piston speed on line 28, and for the total pressure on line 98. This power is calculated for the entire steam pressure down to the zero of pressure, and is the whole power developed by the steam in the small cylinders produced by the weight of water on line 22. It excludes the power required to overcome the cushioning, because the weight of steam cushioned is precisely the weight that is required to produce the power which overcomes the cushioning. If the power required to overcome the cushioning be included in the total power developed by the cylinders, then the weight of steam cushioned per hour must be added to the weight on line 22 for the total weight of steam producing this new total horses-power. Of course, the result would be the same as regards the economic performance of the engine let whichever method be used, but the one adopted by the writer has the advantage of greater simplicity.

Lines 140 and 141 contain respectively the number of indicated and of net horses-power developed in the intermediate cylinders. They are calculated for the entire net area of the pistons of these

cylinders, for the piston speed on line 28, and for the indicated and net pressures on lines 107 and 108.

Line 142 contains the number of total horses-power developed in the intermediate cylinders exclusive of the power required to overcome the cushioning. This power is calculated for the entire steam pressure down to the zero of pressure, line 109, and is the whole power developed by the steam in the intermediate cylinders corresponding to the weight of water on line 22. It excludes the power required to overcome the cushioning. It is calculated for the piston speed on line 28, and for what remains of the net area of the pistons of the intermediate cylinders after subtracting the net area of the pistons of the small cylinders. This subtraction is necessary because the power due to the total pressure down to zero, line 109, for the portion of the net area of the pistons of the intermediate cylinders equal to the net area of the pistons of the small cylinders, has been already included in the total horses-power, line 138, developed by the pistons of the small cylinders.

Line 143 contains the number of horses-power developed by the expanding steam in the intermediate cylinders, including the power required to cushion the back pressure steam in those cylinders. Were there no cushioning, this power would be the same as that on line 142, because all the power developed in the intermediate cylinders is produced by expanding steam, previous to, as well as subsequent to, the closing of the cut-off valves on those cylinders. The power, therefore, on line 143 exceeds the power on line 142 by the power required to cushion the back pressure in the intermediate cylinders. The reason for including the power expanded in this cushioning, is that the expanding steam really overcomes it, and a knowledge of the whole power effect of that steam is necessary in order to calculate the number of units of heat transmuted from the mean latent heat of the expanding steam into the work done by that steam. As the expanding steam in this connection has no relation to economic performance, only the whole power developed by it is required to be known, irrespective of the water weight on line 22.

Line 144 contains the number of indicated horses-power de-

veloped in the large cylinders, calculated for the entire net area of their pistons, for the piston speed on line 28, and for the indicated pressure on line 119.

Line 145 contains the number of net horses-power developed in the large cylinders, calculated for the entire net area of their pistons, for the piston speed on line 28, and for the net pressure on line 120.

Line 146 contains the number of total horses-power developed in the large cylinders exclusive of the power required to overcome the cushioning. This power is calculated for the entire steam pressure down to the zero of pressure, line 121, and is the whole power developed by the steam in the large cylinders corresponding to the weight of water on line 22. It excludes the power required to overcome the cushioning. It is calculated for the piston speed on line 28, and for what remains of the net area of the pistons of the large cylinders after subtracting the net area of the pistons of the intermediate cylinders. This subtraction is necessary because the power due to the total pressure down to zero, line 121, for the portion of the net area of the pistons of the large cylinders equal to the net area of the pistons of the intermediate cylinders, has been already included in the total horses-power, line 142, developed by the pistons of the intermediate cylinders.

Line 147 contains the number of horses-power developed by the expanding steam in the large cylinders, including the power required to cushion the back pressure steam in those cylinders. Were there no cushioning, this power would be the same as that on line 146, because all the power developed in the large cylinders is produced by expanding steam, previous to, as well as subsequent to, the closing of the cut-off valves on those cylinders. The power, therefore, on line 147 exceeds the power on line 146 by the power required to cushion the back pressure in the large cylinders. The power on line 147 is needed for the calculation of the number of units of heat transmuted into the work done in the large cylinders by the expanding steam, and has no reference to the weight of water on line 22. It is calculated for the pressure on line 124, for the piston speed on line

28, and for what remains of the net areas of the pistons of the large cylinders after subtraction of the net areas of the pistons of the intermediate cylinders.

The number of total horses power developed in the small cylinder, *including* the power required to overcome the cushioning, was 318.983, adding to which the number of total horses-power developed in the intermediate cylinder, including the power required to overcome the cushioning, 212.796, line 143, and the number of total horses-power developed in the large cylinder, including the power required to overcome the cushioning, 160.298, line 147, there is obtained the aggregate 692.077 horses-power. Of this sum 515.936 horses-power, line 151, were developed by the expanding steam alone, so that the latter quantity was $(\frac{515.936}{692.077} =)$ 0.745 of the total horses-power developed by the engine, including the power to overcome the cushioning. These latter determinations have no reference to the quantity of feed water consumed on line 22. They simply give the relation of the absolute total work done in the engine, to the portion of that work done by the expanding steam. The assumption is made that the work done by the steam in the small cylinder previous to the closing of the cut-off valve is work done by non-expanded steam. This assumption is erroneous, because the steam was, and necessarily always has to be expanded from the commencement of the stroke of the piston to the point of cutting off. This expansion may be insignificant or considerable according to the proportions and conditions of the mechanism; to calculate the quantity of work done by it is difficult, and, in the case of the present experiment, not necessary, because it was very small, the difference between the initial pressure in the small cylinder (line 90), and the mean pressure between the initial and the cut-off pressure (line 99) being only $(128.5 - 128.129 =)$ 0.371 pound per square inch, a negligible quantity.

The quantity on line 148 is the sum of the quantities on lines 136, 140 and 144.

The quantity on line 149 is the sum of the quantities on lines 137, 141 and 145.

The quantity on line 150 is the sum of the quantities on lines 138, 142 and 146.

The quantity on line 151 is the sum of the quantities on lines 139, 143 and 147.

The quantity on line 152 is the difference between the quantities on lines 149 and 148. This difference is the number of horses-power due to the entire net area of the pistons of the large cylinders, the piston speed on line 28, and the difference between the pressures on lines 125 and 126, namely, 1.077 pounds per square inch, which is assumed as the equivalent pressure required on the pistons of those cylinders to work the engines, *per se*, or unloaded.

The quantity on line 153 is the difference between the quantities on lines 149 and 132.

The quantity on line 154 is the difference between the quantities on lines 150 and 148.

ECONOMIC RESULTS.

The quantities on lines 155, 156, 157 and 158, are the quotients, respectively, of the quantity on line 22 divided by the quantities on lines 148, 149, 150 and 132.

The quantities on lines 159, 160, 161 and 162, are the products, respectively, of the multiplication of the quantities on lines 155, 156, 157 and 158, by the quantity on line 24.

Assuming 789.25 foot-pounds of work as the mechanical equivalent of one Fahrenheit unit of heat; and assuming the horse-power to be 33,000 pounds raised one foot high per minute, or $(33{,}000 \times 60 =)$ 1,980,000 foot-pounds of work per hour, the horse-power per hour is equal to $\left(\frac{1{,}980{,}000}{789.25} =\right)$ 2508.711 Fahrenheit units of heat, and if the entire *vis viva* of the steam could be transmuted into mass work, a horse-power developed by the engine would be obtained by the expenditure of this number of Fahrenheit units of heat. Now the quantities on lines 159, 160, 161 and 162, are the number of Fahrenheit units of heat experimentally expended per hour per indicated, net and total horse-power developed by the engines, and per horse-power applied by the pumps to the net pumping, consequently, the quotients of

the division of 2,508.711 by the number of Fahrenheit units of heat respectively on the lines 159, 160, 161 and 162, will give the fraction of the heat imparted in the boilers to the feed water, that has been experimentally utilized or transmuted into the different horse-powers. These quotients are the quantities on lines 163, 164, 165 and 166.

The true measure of the efficiency of the experimental engines as transmutors of the heat imparted in the boilers to the feed water, into mass *vis viva* or mechanical work, is given by the quantity on line 160, from which is seen that 19 per centum of this heat was transmuted into mass *vis viva*. Of course this fraction was not realized commercially, because of the unavoidably imperfect design of the engines, the imperfect liquefaction of the steam in the condenser, and the various frictions, etc., already pointed out, to be overcome, which, in the aggregate, reduce the efficiency from 19 to 14 per centum, line 166, a huge difference which the efforts of engineers should be directed to diminish.

Weight of Steam Present in the Cylinders, per Indicator.—To calculate the quantity on line 167, the aggregate number of cubic feet in the clearance, steam passage, and portion of the small cylinder up to the point of cutting off the steam, is multiplied by the number of strokes of the piston made per hour and by the weight, in fractions of a pound, of a cubic foot of steam of the pressure at the point of cutting off. From this last product is subtracted a quantity obtained by multiplying the aggregate number of cubic feet in the clearance, steam passage, and portion of the small cylinder included between the end of the stroke of the piston of that cylinder and the point at which the cushioning of the steam began, by the number of strokes of the piston made per hour, and by the weight, in fractions of a pound, of a cubic foot of steam of the pressure at the point where the cushioning began. The remainder is the quantity on line 167, and is all the steam that is accounted for by the indicator at the point of cutting off in the small cylinder, the heat transmuted into the power developed by the steam up to that point being supplied directly by the furnace and not taken from the latent heat of the steam itself.

The calculation of the quantity on line 168 is made in a similar manner to the calculation of the quantity on line 167, substituting for the space in cubic feet between the steam valve and the point of cutting off, the space between the steam valve and the end of the stroke of the piston, and for the weight in fractions of a pound per cubic foot of steam of the pressure at the point of cutting off, the weight in pounds per cubic foot of steam of the pressure at the end of the stroke of the piston. The subtrahend will be the same as in the case of the quantity on line 167. The quantity on line 168 is not the entire weight of steam that can be accounted for by the indicator, as it does not include the weight of steam liquefied to furnish the heat transmuted into the power developed by the steam between the point at which the cut-off valve closed and the end of the stroke of the piston; that is to say, the power developed by the expanding steam alone.

The quantity on line 169 is calculated by first ascertaining the number of foot-pounds of work done per hour equivalent to the number of horses-power (line 139) developed by the expanding steam alone in the small cylinder, that is, multiplying this power by 33,000 and by 60, the number of minutes in an hour; then dividing this product by 789¼, the number of foot-pounds of work equivalent to one Fahrenheit unit of heat, to ascertain the number of Fahrenheit units of heat equivalent to the foot-pounds of work done by the expanding steam; finally, the last quotient is divided by the latent heat of the mean pressure (line 100) of the expanding steam to obtain the number of pounds weight of steam required to be liquefied in order to furnish the heat transmuted into the work done by the expanding steam alone. This last quantity is the quantity on line 169. It is a quantity accounted for by the indicator, and, when added to the quantity on line 167, the sum is the number of pounds weight of steam accounted for per hour in the small cylinder at the end of the stroke of its piston by the indicator, line 170.

The quantity on line 171 is calculated in the same manner as the quantity on line 167, substitution being made of the corresponding data.

All the work done in the intermediate cylinder is done by ex-

panding steam, a fact not affected by the closing of the cut off on that cylinder. The cutting off of the steam in cylinders succeeding the small cylinder affects the problem to the extent of increasing, relatively, the capacity of the receivers between the cylinders, so as to lessen to a certain degree the "drop" and "gap" between their indicator diagrams, to regulate the back pressures in the cylinders so as to equalize approximately the indicated powers developed in them, and to enable the steam due to the revaporization of the water of liquefaction in the preceding cylinders to be used expansively in the succeeding cylinders. In comparing the back pressure of a preceding with the steam pressure of a succeeding cylinder, the back pressure to be considered is the absolute back pressure exclusive of the cushioning; and the steam pressure to be considered is the absolute pressure between the commencement of the stroke of the piston and the point of cutting off.

The quantity on line 172 is the sum of a quantity calculated in the same manner (for the power on line 143) as the quantity on line 169, substitution being made of the corresponding data, and of the quantity on line 169; the quantity on line 172 being the number of pounds weight of steam liquefied to furnish the heat transmuted into the power (calculated for pressure down to zero) developed by the expanding steam from the closing of the cut-off valve on the small cylinder to the end of the stroke of the piston of the intermediate cylinder.

The quantity on line 173 is the sum of the quantities on lines 171 and 172, and is the number of pounds weight of steam accounted for by the indicator per hour at the end of the stroke of the piston of the intermediate cylinder.

The quantities on lines 174, 175 and 176 are calculated in the same manner as the respective quantities on lines 171, 172 and 173, substitution being made of the corresponding data.

Of course, all the calculations in this paper, like all calculations which involve physical measurement, are subject to corrections, many of which, however, are slight enough to be negligible, and most of which neutralize each other, that is to say, some increase and some decrease the result to about an

equal extent. The diameter of the cylinders, and the length of the cylinders (the latter involving variations in the waste spaces at the cylinder ends), are from measurements made with the metal at the atmospheric temperatures; but when the engine is in operation, the temperature of the cylinders and other parts is very much greater, and must cause a corresponding increase of the dimensions; moreover, neither the extreme nor the mean temperature of the cylinder metal, etc., are known. Even the length of the stroke of the piston, though controlled by the crank, does not remain constant, the more or less slackness of the brasses of the connecting rod, and the variable temperature of the piston rod during a stroke, affect the length of the stroke. There is, on the other hand, variation in the strength of the springs of the indicator, which becomes weaker with increase of temperature, and *vice versa*, so that the scale of the spring can only be correct for the temperature at which it was adjusted, and must be erroneous at all other temperatures, and the temperaature must be in continuous variation between certain limits during the time a diagram is taken. Here then is an instance in which the errors of two variables (the temperature of the cylinder and the temperature of the indicator) more or less neutralize each other. If the scale of the spring be adjusted for an imagined mean temperature when the indicator is in use, it will show too much pressure when the temperature is greater and too little when the temperature is less. The inertia of the moving parts of the indicator cause error, also, the frictions of the piston, and of the pencil of the indicator, and of the indicator mechanism. Strings stretch, thus producing an everyway erroneous diagram, and the stretching is not uniform, depending wholly on the pull accidentally or momentarily acting on the string. The very paper on which the diagram is traced varies in dimensions (the diagram dimensions varying accordingly) corresponding to the hydroscopic condition of the atmosphere or rather of the particular portion of the atmosphere surrounding the indicator when the diagram was taken. The moister the air the larger the area of the diagram will be, and *vice versa*, so that there is always the error due to the difference between the dimensions of the diagram when taken from the cylinder and

its dimensions when measured sometime afterwards in the comparatively dry air of an office. The air surrounding the indicator in the engine room of a steamer, saturated with the steam leaking from the stuffing boxes and various cocks of the engine, particularly the indicator cocks, and derived from the vaporization of water always in the bilge of the vessel by the high temperature of the engine room, must always be greatly more humid than the air of an office, so that a considerable error can thus result apparently decreasing the power of the engine.

As the indicator must be connected with the interior of the cylinder by means of a pipe more or less long, the steam pressure on the indicator diagram cannot possibly be as great as the corresponding pressure in the cylinder, and the back pressure or exhaust pressure on the diagram cannot possibly be as little as the corresponding back pressure in the cylinder, so that the diagram gives too small an indicated pressure at every ordinate, both for the steam and the exhaust pressures. The steam cannot get from the cylinder into the indicator unless the pressure in the latter is less than the pressure in the former; nor can the exhaust steam get out of the indicator into the cylinder unless the pressure in the indicator is greater than in the cylinder. Let the area of the indicator pipe be as large as it may, and let the length of this pipe be as short as it may, the corresponding pressures in the cylinder and in the indicator cannot be equalized, though the difference may thus be made quite small. It will become large just in proportion to the smallness of the area and the greatness of the length of the pipe, and to the rapidity of the revolutions of the engine.

The inner surfaces of the steam cylinders are covered with a more or less thick coating of water of steam liquefaction, which, though not decreasing the power developed by the cylinder, decreases the bulk of steam contained in it in proportion to the decrease that the water causes in the capacity of the cylinder. This decrease interferes with the accuracy of the calculations of the quantities of steam accounted for in the cylinder by the indicator; moreover, the water of molecular liquefaction suspended in the steam in the form of mist or fog, likewise interferes with

the accuracy of such calculations. The foregoing considerations are patent qualitative facts, but the corresponding quantitative effects cannot be ascertained. As many, and probably most of these errors in data correct each other, they, in the aggregate, affect the final result to only a negligible extent absolutely, and to scarcely any extent relatively, as they apply about equally to all cylinders and to all indicators.

The weight of steam liquefied by the interaction of the heat of the steam in the cylinder with the metal of the cylinder, and also the weight of steam molecularly liquefied to furnish the heat transmuted into the power developed by the expanding steam alone, are not only calculable quantities, but have been calculated and will be found in the following table. There would have been required, additionally, only the division of these quantities by the weight in pounds per cubic foot of water of the temperature of the steam in contact therewith, to obtain the quantities to be subtracted from those on lines 167, 168, 171 and 174, in order to make the corrections required for the water present in the cylinder. The corrections thus obtained, however, are insignificant enough to be negligible. Unless *all* the corrections can be made, all had better be omitted, and reliance placed on the correction of errors where a moiety increases and a moiety decreases the final result; consequently, the writer in his many experimental investigations, has never applied any of these corrections, for they would only have made what was not quite satisfactory still more unsatisfactory.

Differences between the Weight of Steam entering the Small Cylinders and the Weights of Steam accounted for in those and their Succeeding Cylinders by the Indicator.—The quantity on line 177 is the remainder of the quantity on line 23 after subtraction of the quantity on line 167. The quantity on line 177 is the total number of pounds weight of steam liquefied per hour in the small cylinders by the interaction of the heat in the steam with the metal of these cylinders. It is wholly a surface liquefaction notwithstanding the heat received by the surfaces from the steam in the jackets, and the water is deposited in drops or layers uniformly on all the interior surfaces of the cylinders. This

quantity is, of course, exclusive of any loss of heat by external radiation, for such loss is prevented by the jackets.

The quantity on line 178 is the fraction which the quantity on line 177 is of the quantity on line 23. The quantity on line 179 is the remainder of the quantity on line 23 after subtraction of the quantity on line 170. The quantity on line 179 is the total number of pounds weight of steam remaining liquefied per hour in the small cylinders at the end of the stroke of their pistons from the interaction of the heat in the steam with the metal of these cylinders. The subtraction of the quantity on line 179 from the quantity on line 178, gives the number of pounds of water of liquefaction revaporized in the small cylinders per hour between the point of cutting off the steam and the end of the stroke of the piston. This revaporization was only made possible by the reduction of the pressure due to the expansion of the steam, and was made at the expense partly of the contained heat in the water, and partly at the expense of the heat in the metal, and partly at the expense of the heat in the steam in the jackets. It amounted to $\left(\frac{1,737.952-822.950}{1,737.952}=\right)$ 0.5265 of the water of liquefaction present in the small cylinders at the point of cutting off, the remaining 0.4735 being revaporized during the return or expansion stroke of the piston and the resulting steam passed into the intermediate cylinders for the production of pressure on their pistons. Had there been no surface liquefaction of steam in the small cylinders, the whole of the 1,737.952 pounds weight of steam liquefied per hour during the portion of the stroke of the pistons previous to the closing of the cut-off valve, would have produced its corresponding power. This water of liquefaction had been once vaporized in the boiler and the steam from it had produced no power up to the point of cutting off. Subsequent to the point of cutting off, the revaporization of 0.5265 of this water produced power during the portion of the stroke of the pistons between the point of cutting off and the end of their stroke, and the remaining 0.4735 of it passed to the intermediate cylinders, producing power pressure upon their pistons the same as produced by the steam exhausted as such from the small cylinders. The revaporization of the 0.5265 part

of the water of liquefaction between the point of cutting off and the end of the stroke of the small cylinders, produced but very little power relatively to what it would have produced had it been vaporized in the boiler and used expansively from the point of cutting off to the end of the stroke of the piston, because, subsequent to the point of cutting off, the whole of the space between the steam valve and any point at which part of the water of liquefaction might be revaporized, acted as waste space to the steam thus generated, its pressure being distributed uselessly through that space, the steam being expanded backwards as well as forwards, so that really the power produced by the steam of the revaporized water in the small cylinders was of not much benefit absolutely, and still less relatively when considered with reference to the power that might have been produced from it, had it been vaporized in the boiler. The steam produced from the 0.4735 part of this water during the exhaust stroke of the pistons of the small cylinders passed into the receivers, and was properly used upon the pistons of the intermediate cylinders during their entire stroke. The 0.4735 part produces the same power effect as though it had been vaporized in the boiler at the receiver pressure, delivered directly into the receiver, and used throughout the succeeding cylinders. The power thus produced is, nevertheless, much inferior to what it would have been had the water been vaporized in the boiler at the boiler pressure and then delivered into the small cylinders.

Obviously, the weight of steam liquefied in the small cylinders by the interaction of its heat with the metal of the cylinders, was twice evaporated, once in the boiler and once in the small cylinders, and that scarcely any power was derived from it in those cylinders. This weight of liquefied steam was much less than it would have been had there been no jackets to the cylinder; it was roundly as much less as the weight of the water of liquefaction drawn per hour from the jackets of the small cylinders, the steam liquefied in the jackets having been evaporated but once and, by having prevented the equivalent liquefaction in the cylinders, prevented the second vaporization there. Hence, the economic effect of steam jacketing the small cylinders was,

as regards those cylinders alone, 0.019 (line 11) of the heat imparted to the feed water in the boiler, or, say, 2 per centum.

All the steam that entered the small cylinders, enters the receivers between them and the intermediate cylinders with the exception of the steam molecularly liquefied to furnish the heat transmuted into the power developed by the expanding steam alone in the small cylinders. The water of this liquefaction being suspended in the steam in the state of the water in fog or mist, and excessively comminuted, is swept as such with and by the exhaust steam from the small cylinders into the receivers between them and the intermediate cylinders. The weight of steam that as such entered the receivers per hour, was (6,598.833 pounds, line 23, less 399.780 pounds, line 169) 6,199.053 pounds. In the receivers there is more liquefaction of steam, due to external radiation. Some of this water will be revaporized in the receivers under the fluctuations of pressure therein, by its own contained heat and by the heat of the metal; the remainder will, from time to time as the accumulations become great enough, be driven into the intermediate cylinders as water along with the steam. But although the intermediate cylinders will, therefore, receive a decidedly less weight of steam as such than the small cylinders, they will receive the same aggregate weight of steam and water.

The quantity on line 180 is the fraction which the quantity on line 179 is of the quantity on line 23.

The quantity on line 181 is the remainder of the quantity on line 23 after subtraction of the quantity on line 173. The quantity on line 181 is the total number of pounds weight of steam remaining liquefied per hour in the intermediate cylinders at the end of the stroke of their pistons. All the water, however caused, which entered the intermediate cylinders with the steam from the receivers, had then been converted into steam together with a large portion of the water of molecular liquefaction due to the total power, including overcoming the back pressure, developed in these cylinders, for all the pressure in them was produced by expanding steam. The weight due to this molecular liquefaction per hour was (963.371, line 172, minus 399.780 line 169=) 563.591

pounds, all of which except the 77.301 pounds on line 181 was vaporized. Thus of the entire weight of steam which entered the small cylinder per hour, all, except one and one-sixth per centum (line 182) was present as steam at the end of the stroke of the pistons of the intermediate cylinders; this small fraction, however, maintained the steam in the saturated state. The production of this condition was due entirely to the heat passed from the steam jackets into the cylinders, and they must have acted very efficiently.

Of the 6,598.833 pounds weight of steam and water, line 23, which entered per hour the intermediate cylinders, the jacket heat could produce no economic effect upon the molecular liquefaction 963.371 pounds, line 172, nor upon the 77.301 pounds of surface liquefaction, line 181, making a total of 1,040.672 pounds per hour, leaving as the weight of steam subject to the economic action of the jacket heat (6,598.833—1,040.672=) 5,558.161 pounds per hour, or the fraction 0.8423, because any portion of this weight of steam could be both liquefied and its water of liquefaction revaporized in the cylinder without doing any work of consequence, so that it would be twice evaporated, once in the boiler and afterwards in the cylinder. Now, by the action of the jacket heat, a weight of steam equivalent in heat to the weight liquefied in the cylinder, has only to be evaporated once (in the boiler), thus saving the heat of one evaporation, namely, the one in the cylinder. Hence, the economic value of the jackets as regards the intermediate cylinders, will be represented by the fraction of steam liquefied in the jackets, line 12, multiplied by the fraction 0.8423 above obtained, or, by (0.040294×0.8423=) 0.03394; that is to say, the economy produced by the jackets on the intermediate cylinders was, roundly 3.4 per centum of the heat imparted to the feed water in the boilers, adding to which the, say, 2 per centum of such economy given, as above shown, by the jackets of the small cylinders, there results for the aggregate economy of the jackets of the small and of the intermediate cylinders, 5.4 per centum of the heat imparted to the feed water in the boilers. If some allowance be made of the power produced by the revaporization of the water of liquefaction in the cylinders

between the point of cutting off and the end of the stroke of the pistons, this percentage can be reduced to 5.

Owing to the fact that there were no liquefactions of steam in the large cylinders, and that the steam from the intermediate cylinders entered the large ones sensibly free of water, the jackets on the large cylinders produced no economic effect, notwithstanding the considerable weight of steam liquefied in them, this liquefaction being, under the circumstances, a mere transfer of equal quantities of heat from jacket steam to cylinder steam, producing superheating of the latter but no economic effect, the production of such effect being restricted to the prevention by the jacket heat of surface liquefaction of steam in the cylinder.

The absence of economy by the use of jackets on the large cylinders, limits the jacket economy of the experimental engine to the above 5 per centum. The result thus obtained coincides closely with the results of comparative experiments on triple expansion engines using steam with and without jacketing. So small an economy is difficult to determine experimentally because the experimental errors may easily, and generally do, amount to that per centum as extremes. In the case where only the barrels or sides of the cylinders are steam jacketed, and not the whole of them because the valve chests cover a considerable fraction of their area, the gain by steam jacketing is, of course, still less, and may easily fall to 2 per centum, steam of the boiler pressure being used in the jackets.

The quantity on line 182 is the fraction which the quantity on line 181 is of the quantity on line 23.

The quantity on line 183 is the remainder of the quantity on line 176 after subtraction of the quantity on line 23, and shows that the steam in the large cylinders at the end of the stroke of their pistons was superheated sufficiently to make the pressure there, considered as the pressure of saturated steam, great enough to show an excess of weight of steam of 162.028 pounds per hour over the quantity entering the small cylinders per hour, line 23.

The quantity on line 184 is the fraction which the quantity on line 183 is of the quantity on line 23.

TABLE CONTAINING THE DATA AND RESULTS OF THE TWO EXPERIMENTS MADE BY MR. MICHAEL LONGRIDGE SIMULTANEOUSLY ON THE TWO TRIPLE EXPANSION, DIRECT ACTING, STEAM JACKETED, SURFACE CONDENSING, VERTICAL PUMPING ENGINES AT THE EAST LONDON WATER WORKS, LEA BRIDGE, ENGLAND.

Date of the Experiment.	1893. 1st March.	1893. 9th March.
TOTAL QUANTITIES.		
1. Duration of the experiment in consecutive minutes	406.	360.
2. Total number of double strokes made by the pistons of the North Engine	8,729.	7,925.
3. Total number of double strokes made by the pistons of the Middle Engine	8,625.	7,858.
4. Total number of pounds of feed water vaporized in the boilers	48,055.	44,870.
5. Total number of pounds of water of steam liquefaction drawn from the jackets of the small cylinders	930.	862.
6. Total number of pounds of water of steam liquefaction drawn from the jackets of the intermediate cylinders	2,180.	1,808.
7. Total number of pounds of water of steam liquefaction drawn from the jackets of the large cylinders	2,757.	2,421.
8. Total number of pounds of water of steam liquefaction drawn from all the jackets of all the cylinders	5,867.	5,091.
9. Total number of pounds of water of steam liquefaction drawn from the steam pipe	241.	186.
10. Total number of pounds weight of steam that entered the small cylinders	41,947.	39,593.
11. Fraction which the water of steam liquefaction drawn from the jackets of the small cylinders is of the feed water vaporized in the boilers	0.019	0.019
12. Fraction which the water of steam liquefaction drawn from the jackets of the intermediate cylinders is of the feed water vaporized in the boilers	0.045	0.040
13. Fraction which the water of steam liquefaction drawn from the jackets of the large cylinders is of the feed water vaporized in the boilers	0.057	0.054
14. Fraction which the water of steam liquefaction drawn from all the jackets of all the cylinders is of the feed water vaporized in the boilers	0.122	0.113

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15. Fraction which the water of steam liquefaction drawn from the steam pipe is of the feed water vaporized in the boilers	0.005	0.004
16. Fraction which the weight of steam that entered the small cylinders is of the weight of steam evaporated in the boilers	0.873	0.882
17. Total number of pounds of water of steam liquefaction produced in the jackets of the small cylinders by their external radiation of heat	433.067	384.000
18. Total number of pounds of water of steam liquefaction produced in the jackets of the intermediate cylinders by their external radiation of heat	663.133	588.000
19. Total number of pounds of water of steam liquefaction produced in the jackets of the large cylinders by their external radiation of heat	1 745.800	1,548.000
20. Total number of pounds of water of steam liquefaction produced in all the jackets of all the cylinders by their external radiation of heat,	2 842.000	2,520.000
21. Fraction which the water of steam liquefaction produced in all the jackets of all the cylinders by their external radiation of heat, is of the weight of water vaporized in the boilers	0 059	0.056
ENGINES.		
22. Number of pounds of feed water vaporized in the boilers per hour	7,101.724	7,478.333
23. Number of pounds weight of steam that entered the small cylinders per hour	6,199.064	6,598.833
24. Number of Fahrenheit units of heat imparted to each pound of water vaporized in the boilers, supposing the temperature of the feed water to be 3.9 degrees Fahrenheit less than what is due to the condenser temperature of 72 degrees and to the temperatures of the waters of steam liquefaction drawn from all the jackets of all the cylinders and to the temperature of the water of steam liquefaction drawn from the steam pipe, supposing these waters of steam liquefaction to be used as part of the feed water as they would be in the regular service of the engines	1,117.539	1,120.582
25. Mean number of double strokes made per minute by the pistons of the North Engine	21.500	22.014
26. Mean number of double strokes made per minute by the pistons of the Middle Engine	21.244	21.828

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27. Mean number of double strokes made per minute by the pistons of both engines	21.372	21.920
28. Mean speed of the pistons of both engines in feet per minute	170.975	175.367
29. Steam pressure in the boilers in pounds per square inch above the atmosphere	120.0	120.0
30. Steam pressure in the boilers in pounds per square inch above zero	134.6	134.8
31. Steam pressure in the steam jackets of the small cylinders, in pounds per square inch above the atmosphere	116.0	116.0
32. Steam pressure in the steam jackets of the small cylinders, in pounds per square inch above zero	130.6	130.8
33. Steam pressure in the steam jackets of the intermediate cylinders, in pounds per square inch above the atmosphere	115.0	114.0
34. Steam pressure in the steam jackets of the intermediate cylinders, in pounds per square inch above zero	129.6	128.8
35. Steam pressure in the steam jackets of the large cylinders, in pounds per square inch above the atmosphere	113.0	113.0
36. Steam pressure in the steam jackets of the large cylinders, in pounds per square inch above zero,	127.6	127.8
37. Pressure of the atmosphere in pounds per square inch	14.6	14.8
38. Pressure in the condenser below the atmospheric pressure in pounds per square inch by gauge...	14.0	14.26
39. Pressure in the condenser in pounds per square inch above zero	0.6	0 54
40. Fraction completed of the stroke of the pistons of the small cylinders when the steam was cut off,	0.40	0.425
41. Fraction completed of the stroke of the pistons of the intermediate cylinders when the steam was cut off	0.40	0.44
42. Fraction completed of the stroke of the pistons of the large cylinders when the steam was cut off,	0.55	0.55
43. Fraction completed of the stroke of the pistons of the small cylinders when the steam was released	0.97	0.97
44. Fraction completed of the stroke of the pistons of the intermediate cylinders when the steam was released	0.97	0.97
45. Fraction completed of the stroke of the pistons of the large cylinders when the steam was released,	0.90	0 90

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46. Fraction completed of the exhaust stroke of the pistons of the small cylinders when the cushioning of the back pressure began	0.15	0.15
47. Fraction completed of the exhaust stroke of the pistons of the intermediate cylinders when the cushioning of the back pressure began	0.125	0.125
48. Fraction completed of the exhaust stroke of the pistons of the large cylinders when the cushioning of the back pressure began	0.25	0.25
49. Number of times the steam was expanded in the small cylinders	2.341	2.217
50. Number of times the steam was expanded in the engines	19.412	18.385
TEMPERATURES IN DEGREES FAHRENHEIT.		
51. Temperature of the steam in the boilers	349.8	349.9
52. Temperature of the steam in the jackets of the small cylinders	347.47	347.59
53. Temperature of the steam in the jackets of the intermediate cylinders	346.88	346.41
54. Temperature of the steam in the jackets of the large cylinders	345.69	345.81
55. Mean temperature of the steam in the small cylinders, including back pressure	.....	304.6
56. Mean temperature of the steam in the intermediate cylinders, including back pressure	.....	232.1
57. Mean temperature of the steam in the large cylinders, including back pressure	.....	155.8
58. Difference between the temperature of the steam in the jackets of the small cylinders, and the mean temperature of the steam in those cylinders	.....	45.3
59. Difference between the temperature of the steam in the jackets of the intermediate cylinders, and the mean temperature of the steam in those cylinders	.....	117.8
60. Difference between the temperature of the steam in the jackets of the large cylinders, and the mean temperature of the steam in those cylinders	.....	194.1
61. Temperature of the feed water entering the "economizer" from the tank	55.	62.3
62. Temperature of the feed water entering the boilers from the "economizer"	151.	131.6
63. Temperature of the water and steam in the condenser	72.	72.

	1 Mar., '93.	9 Mar., '93.
64. Temperature which the feed water entering the "economizer" would have had, had it been supplied from the hot well of the air pump and from all the jackets of all the cylinders, together with the drainage from the steam pipe, and had it suffered no loss of temperature by external radiation of heat during its transits	106.9	104.3
65. Temperature of the feed water entering the "economizer," from which the total heat of vaporization is calculated, being the temperature on the immediately preceding line less 3.9 degrees allowed for the loss due to external radiation from the feed pipes, etc	103.0	100.4
HEAT.		
66. Number of Fahrenheit units of heat imparted per hour to the feed water vaporized in the boilers,	7,936,455.388	8,380,086.830
67. Number of Fahrenheit units of heat imparted per hour to that portion of the feed water vaporized in the boilers which furnished the steam entering the small cylinders	6,927,697.309	7,394,534.830
68. Number of Fahrenheit units of heat externally radiated per hour by the steam in the jackets of the small cylinders	55,579.686	55,574.176
69. Number of Fahrenheit units of heat passed per hour from the steam in the jackets of the small cylinders into the steam within those cylinders,	63,776.321	69,178.271
70. Number of Fahrenheit units of heat passed per hour from the steam in the jackets of the small cylinders into the steam within those cylinders, per square foot of jacketed surface	1,080.955	1,172 513
71. Number of Fahrenheit units of heat externally radiated per hour by the steam in the jackets of the intermediate cylinders	85,148.580	85,182.466
72. Number of Fahrenheit units of heat passed per hour from the steam in the jackets of the intermediate cylinders into the steam within those cylinders	194,770.849	176,739.131
73. Number of Fahrenheit units of heat passed per hour from the steam in the jackets of the intermediate cylinders into the steam within those cylinders, per square foot of jacketed surface	1,913.269	1,736.141
74. Number of Fahrenheit units of heat externally radiated per hour by the steam in the jackets of the large cylinders	224,391.369	224,368.790

	1 Mar., '93.	9 Mar., '93.
75. Number of Fahrenheit units of heat passed per hour from the steam in the jackets of the large cylinders into the steam within those cylinders,	129,971.700	126,533 562
76. Number of Fahrenheit units of heat passed per hour from the steam in the jackets of the large cylinders into the steam within those cylinders per square foot of jacketed surface.....	738.475	718.941
77. Number of Fahrenheit units of heat passed per hour from the steam in the jackets of the small cylinders into the steam within those cylinders per square foot of jacketed surface per degree difference of temperature between the temperature of the jacket steam and the mean temperature of the steam within the small cylinders..	.....	25 883
78. Number of Fahrenheit units of heat passed per hour from the steam in the jackets of the intermediate cylinders into the steam within those cylinders per square foot of jacketed surface per degree difference of temperature between the temperature of the jacket steam and the mean temperature of the steam within the intermediate cylinders.....	.....	14.733
79. Number of Fahrenheit units of heat passed per hour from the steam in the jackets of the large cylinders into the steam within those cylinders per square foot of jacketed surface per degree difference of temperature between the temperature of the jacket steam and the mean temperature of the steam within the large cylinders...	.....	3.704
80. Number of Fahrenheit units of heat lost per hour by the steam in the jackets of the small cylinders.....	119.356.007	124,752.447
81. Number of Fahrenheit units of heat lost per hour by the steam in the jackets of the intermediate cylinders	279.919.428	261,921.597
82. Number of Fahrenheit units of heat lost per hour by the steam in the jackets of the large cylinders.....	354.363.069	350,902.352
83. Aggregate number of Fahrenheit units of heat lost per hour by the steam in all the jackets of all the cylinders.....	753,638.505	737.576.396
84. Number of Fahrenheit units of heat imparted per hour to the feed water vaporized in the boilers,	7,936,455.389	8,380,086.830
85. Fraction which the aggregate number of units of heat lost per hour by the steam in all the jackets of all the cylinders is of the number of units		

	1 Mar., '93.	9 Mar., '93.
of heat imparted per hour to the feed water vaporized in the boilers	0.095	0.088
86. Aggregate number of Fahrenheit units of heat exteriorly radiated per hour by the steam in all the jackets of all the cylinders	365,119.634	365,125.432
87. Aggregate number of Fahrenheit units of heat passed per hour from the steam in all the jackets of all the cylinders to the steam within those cylinders	388,518.870	372,450.964
88. Fraction which the aggregate number of units of heat exteriorly radiated per hour by the steam in all the jackets of all the cylinders is of the number of units of heat imparted per hour to the feed water vaporized in the boilers	0.046	0.043
89. Fraction which the aggregate number of units of heat passed per hour from the steam in all the jackets of all the cylinders to the steam within those cylinders, is of the number of units of heat imparted per hour to the feed water vaporized in the boilers	0.049	0.044
STEAM PRESSURES IN THE SMALL CYLINDERS.		
90. Pressure on the pistons of the small cylinders at the commencement of their stroke, in pounds per square inch above zero	.....	128.5
91. Pressure on the pistons of the small cylinders at the point of cutting off the steam, in pounds per square inch above zero	.....	119.5
92. Pressure on the pistons of the small cylinders at the end of their stroke, in pounds per square inch above zero	.....	56.0
93. Mean back pressure, including cushioning, against the pistons of the small cylinders, in pounds per square inch above zero	.....	53.0
94. Mean back pressure, excluding cushioning, against the pistons of the small cylinders, in pounds per square inch above zero	.....	52.5
95. Back pressure against the pistons of the small cylinders at the point where the cushioning began, in pounds per square inch above zero	.....	43.5
96. Mean indicated pressure on the pistons of the small cylinders, in pounds per square inch	.....	45.615
97. Mean net pressure on the pistons of the small cylinders, in pounds per square inch	.....	42.615
98. Mean total pressure, exclusive of cushioning, on the pistons of the small cylinders, in pounds per square inch above zero	.....	90.240

	1 Mar., '93.	9 Mar., '93.
99. Mean pressure on the pistons of the small cylinders, inclusive of cushioning, previous to the point of cutting off, in pounds per square inch above zero.....	.....	128.129
100. Mean pressure on the pistons of the small cylinders of the expanding steam alone after the closing of the cut-off valve, in pounds per square inch above zero.....	.....	76.800

STEAM PRESSURES IN THE INTERMEDIATE CYLINDERS.

101. Pressure on the pistons of the intermediate cylinders at the commencement of their stroke, in pounds per square inch above zero.....	.....	52.50
102. Pressure of the pistons of the intermediate cylinders at the point of cutting off the steam, in pounds per square inch above zero.....	.....	36.75
103. Pressure on the pistons of the intermediate cylinders at the end of their stroke, in pounds per square inch above zero.....	.....	17.75
104. Mean back pressure, including cushioning, against the pistons of the intermediate cylinders, in pounds per square inch above zero.....	.....	13.985
105. Mean back pressure, excluding cushioning, against the pistons of the intermediate cylinders, in pounds per square inch above zero.....	.....	13.271
106. Back pressure against the pistons of the intermediate cylinders, in pounds per square inch above zero.....	.....	12.500
107. Mean indicated pressure on the pistons of the intermediate cylinders, in pounds per square inch,	.....	19.735
108. Mean net pressure on the pistons of the intermediate cylinders, in pounds per square inch.....	.....	18.718
109. Mean total pressure, exclusive of cushioning, on the pistons of the intermediate cylinders, in pounds per square inch above zero.....	.....	31.347
110. Mean pressure on the pistons of the intermediate cylinders, inclusive of cushioning, previous to the point of cutting off, in pounds per square inch above zero	.....	45.947
111. Mean pressure on the pistons of the intermediate cylinders subsequent to the point of cutting off, in pounds per square inch above zero.....	.....	24.113
112. Mean pressure, inclusive of cushioning, on the pistons of the intermediate cylinders of the expanding steam throughout their stroke, in pounds per square inch above zero.....	.....	33.720

STEAM PRESSURES IN THE LARGE CYLINDERS.

	1 Mar., '93.	9 Mar., '93.
113. Pressure on the pistons of the large cylinders at the commencement of their stroke, in pounds per square inch above zero	.....	13.5
114. Pressure on the pistons of the large cylinders at the point of cutting off the steam, in pounds per square inch above zero	.....	9.0
115. Pressure on the pistons of the large cylinders at the end of their stroke, in pounds per square inch above zero	.....	5.5
116. Mean back pressure, including cushioning, against the pistons of the large cylinders, in pounds per square inch above zero	.....	1.91
117. Mean back pressure, excluding cushioning, against the pistons of the large cylinders, in pounds per square inch above zero	.....	1.416
118. Back pressure against the pistons of the large cylinders at the point where the cushioning began, in pounds per square inch above zero	.....	1.5
119. Mean indicated pressure on the pistons of the large cylinders, in pounds per square inch	.....	7.265
120. Mean net pressure on the pistons of the large cylinders, in pounds per square inch	.....	6.906
121. Mean total pressure, exclusive of cushioning, on the pistons of the large cylinders, in pounds per square inch above zero	.....	8.327
122. Mean pressure on the pistons of the large cylinders, inclusive of cushioning, previous to the point of cutting off, in pounds per square inch above zero	.....	11.453
123. Mean pressure on the pistons of the large cylinders subsequent to the point of cutting off, in pounds per square inch above zero	.....	6.39
124. Mean pressure, inclusive of cushioning, on the pistons of the large cylinders of the expanding steam throughout their stroke, in pounds per square inch above zero	.....	9.175

EQUIVALENT STEAM PRESSURES FOR THE LARGE CYLINDERS.

125. Indicated pressure, in pounds per square inch, that would be on the pistons of the large cylinders were the indicated pressure on the pistons of the small cylinders divided by the ratio of the net areas of these pistons and the quotient added to the experimental indicated pressure on

	1 Mar., '93.	9 Mar., '93.
the pistons of the large cylinders; and were the indicated pressures on the pistons of the intermediate cylinders divided by the ratio of the net areas of the pistons of the large cylinders to the net area of the pistons of the intermediate cylinders and the quotient also added to the experimental indicated pressure on the pistons of the large cylinders.....	19.609	19.699
126. Net pressure, in pounds per square inch, that would be on the pistons of the large cylinders were the net pressure on the pistons of the small cylinders divided by the ratio of the net areas of these pistons and the quotient added to the experimental net pressure on the pistons of the large cylinders; and were the net pressure on the pistons of the intermediate cylinders divided by the ratio of the net area of the pistons of the large cylinders to the net area of the pistons of the intermediate cylinders and the quotient also added to the experimental net pressure on the pistons of the large cylinders.....	18.532	18.622
127. Total pressure, in pounds per square inch above zero, that would be on the pistons of the large cylinders to correspond to the number of total horses-power developed by the engines, supposing the entire net areas of the pistons of the large cylinders to be employed.....	.....	23.511
128. Back pressure, in pounds per square inch above zero, that would be against the pistons of the large cylinders, supposing the entire net areas of the pistons of the large cylinders to be employed.....	.....	3.812
129. Back pressure, in pounds per square inch above zero, that would be against the pistons of the large cylinders, for the portion of the stroke of those pistons previous to the commencement of the cushioning.....	.....	5.082
PUMPS.		
130. Pressure, in pounds per square inch per gauge, in the main just outside of the delivery valves of the pumps, including the pressure of the water due to the height between the gauge and the water level in the pump well.....	41.09	41.98
131. Pressure, in pounds per square inch, on the pistons of the large cylinders, equivalent to the		

	1 Mar., '93.	9 Mar., '93.
pressure in the main just outside of the delivery valves of the pumps, supposing the pumps to fill to 0 98 of their capacity.....	16.796	17.160
132. Net horses-power expended in the pumping of the water, that is to say, exclusive of the power required to open the receiving and delivering valves of the pumps, and to force the water through their openings; and to overcome the surface resistance of the suction main and the resistance of its water deflection; supposing the pumps to fill to 0.98 of their capacity..	442.422	462.564
133. Pressure, in pounds per square inch, on the pistons of the large cylinders equivalent to the pressure required to open the receiving and delivering valves of the pumps and to force the water through their openings, and to overcome the friction of the load due to whatever power may have been transmitted through the fly-wheel mechanism, and to overcome the surface resistance of the suction main and the resistance of its water deflection	1.735	1.462
134. The efficiency of the entire mechanism from the steam pistons of the engines, as mechanism, the maximum being unity, as shown by the fraction which the net horses-power expended in the pumping of the water, is of the indicated horses-power developed by the engines.....	0.856	0.869
135. Fraction of the indicated horses-power developed by the engines, expended in working the engines, *per se*, or unloaded, in overcoming the friction of the load passed through the fly-wheel mechanism, in opening the receiving and delivering valves of the pumps and in forcing the water through the openings of the valves, and in overcoming the surface resistance of the suction main; supposing the pumps to fill to 0.98 of their capacity	0.143	0.131
HORSES-POWER.		
136. Indicated horses-power developed in the small cylinders.....	.....	147.548
137. Net horses-power developed in the small cylinders,	.....	137.844
138. Total horses-power developed in the small cylinders, excluding overcoming the cushioning.....	.....	291.894
139. Total horses-power developed in the small cylinders by the expanding steam alone.....	.....	142.841

	1 Mar., '93.	9 Mar., '93.
140. Indicated horses-power developed in the intermediate cylinders	.....	188.377
141. Net horses-power developed in the intermediate cylinders	.....	178.669
142. Total horses-power developed in the intermediate cylinders by the annular surface of their pistons remaining after deducting from the area of those pistons the area of the pistons of the small cylinders, excluding overcoming the cushioning	.....	197.821
143. Total horses-power developed in the intermediate cylinders by the expanding steam acting upon the annular surface of their pistons remaining after deducting from the area of those pistons the area of the pistons of the small cylinders, including overcoming the cushioning	.....	212.796
144. Indicated horses-power developed in the large cylinders	.....	196.275
145. Net horses-power developed in the large cylinders	.....	186.576
146. Total horses-power developed in the large cylinders by the annular surface of their pistons remaining after deducting from the area of those pistons the area of the pistons of the intermediate cylinders, excluding overcoming the cushioning	.....	145.483
147. Total horses-power developed in the large cylinders by the expanding steam acting upon the annular surface of their pistons remaining after deducting from the area of those pistons the area of the pistons of the intermediate cylinders, including overcoming the cushioning	.....	160.298
148. Aggregate indicated horses-power developed by the engines	516.501	532.200
149. Aggregate net horses-power developed by the engine	488.133	503.089
150. Aggregate total horses-power developed by the engines, excluding overcoming the cushioning,	.....	635.197
151. Aggregate total horses-power developed by the expanding steam, including overcoming the cushioning	.....	515.936
152. Aggregate horses-power consumed in working the engines, *per se*, or unloaded	28.368	29.110
153. Aggregate horses-power consumed in overcoming the resistances of all the valves of all the pumps to opening, the resistances of the water to being driven through the openings of these valves,		

	1 Mar., '93.	9 Mar., '93.
and the resistances of the frictions of the load upon the mechanism of the fly-wheel system....	36.682	40.525
154. Aggregate horses-power consumed in overcoming the back pressure against the steam pistons of the engines, excluding overcoming the cushionings.....	.....	102.997
ECONOMIC RESULTS.		
155. Pounds of feed water consumed per hour per indicated horse-power developed by the engines,	13.749	14.052
156. Pounds of feed water consumed per hour per net horse-power developed by the engines...	14.549	14.865
157. Pounds of feed water consumed per hour per total horse-power developed by the engines...	.....	11.773
158. Pounds of feed water consumed per hour per horse-power employed in pumping the net water	15.731	15.865
159. Fahrenheit units of heat consumed per hour per indicated horse-power developed by the engines...	15,365.794	15,746.127
160. Fahrenheit units of heat consumed per hour per net horse-power developed by the engines...	16,258.788	16,657.250
161. Fahrenheit units of heat consumed per hour per total horse-power developed by the engine.....	.....	13,192.888
162. Fahrenheit units of heat consumed per hour per horse-power employed in pumping the net water.....	17,579.890	17,754.246
163. Fraction of the total *vis viva* in the steam transmuted into the work of the indicated horses-power	0.163	0.159
164. Fraction of the total *vis viva* in the steam transmuted into the work of the net horses-power...	0.154	0.151
165. Fraction of the total *vis viva* in the steam transmuted into the work of the total horses-power..	.....	0.190
166. Fraction of the total *vis viva* in the steam transmuted into the work of the horses-power employed in pumping the net water.....	0.143	0.141
WEIGHT OF STEAM PRESENT IN THE CYLINDERS, PER INDICATOR.		
167. Pounds weight of steam per hour in the small cylinders at the point of cutting off the steam.....	.....	4,860.881
168. Pounds weight of steam per hour in the small cylinders at the end of the stroke of their pistons,	.....	5,376.103
169. Pounds weight of steam per hour liquefied in the small cylinders to furnish the heat transmuted		

	1 Mar., '93.	9 Mar., '93.
into the power developed in those cylinders by the expanding steam alone	.....	399 780
170. Sum of the two immediately preceding quantities,	.....	5,775.883
171. Pounds weight of steam per hour in the intermediate cylinders at the end of the stroke of their pistons	.....	5,558.162
172. Pounds weight of steam per hour liquefied in the intermediate and in the small cylinders to furnish the heat transmuted into the power developed in these cylinders by the expanding steam alone	.....	963.371
173. Sum of the two immediately preceding quantities,	.....	6,521.532
174. Pounds weight of steam per hour in the large cylinders at the end of the stroke of their pistons,	.....	5,392 668
175. Pounds weight of steam per hour liquefied in the large, intermediate and small cylinders to furnish the heat transmuted into the power developed in these cylinders by the expanding steam alone	.....	1,368.193
176. Sum of the two immediately preceding quantities,	.....	6,760.861

DIFFERENCES BETWEEN THE WEIGHT OF STEAM ENTERING THE SMALL CYLINDERS AND THE WEIGHTS OF STEAM ACCOUNTED FOR IN THOSE AND THEIR SUCCEEDING CYLINDERS BY THE INDICATOR.

177. Difference in pounds between the weight of steam entering the small cylinders per hour and the weight of steam accounted for per hour in those cylinders by the indicator at the point of cutting off	.....	1,737.952
178. Difference in fractions of the weight of steam entering the small cylinders per hour, between that weight and the weight of steam accounted for per hour by the indicator in the small cylinders at the point of cutting off	.....	0.263
179. Difference in pounds between the weight of steam entering the small cylinders per hour and the weight of steam accounted for per hour in those cylinders by the indicator at the end of the stroke of their pistons	.....	822 950
180. Difference in fractions of the weight of steam entering the small cylinders per hour, between that weight and the weight of steam accounted for per hour by the indicator in the small cylinders at the end of the stroke of their pistons,	.....	0.125

	1 Mar., '93.	9 Mar., '93.
181. Difference in pounds between the weight of steam entering the small cylinders per hour and the weight of steam accounted for per hour in the intermediate cylinders by the indicator at the end of the stroke of their pistons.....	.....	77.301
182. Difference in fractions of the weight of steam entering the small cylinders per hour, between that weight and the weight of steam accounted for per hour by the indicator in the intermediate cylinders at the end of the stroke of their pistons,	.....	0.012
183. Difference in pounds between the weight of steam entering the small cylinders per hour and the weight of steam accounted for per hour in the large cylinders by the indicator at the end of the stroke of their pistons (superheated).....	.....	+162.028
184. Difference in fractions of the weight of steam entering the small cylinders per hour, between that weight and the weight of steam accounted for per hour by the indicator in the large cylinders at the end of the stroke of their pistons (superheated)..	.....	+0.024

LAWS OF SIMILITUDE REGARDING QUESTIONS OF NAVAL CONSTRUCTION.

By Mr. J. A. Normand.

[Translation of a Paper read before L'Association Technique Maritime.]

The theory of similitude, as now established, would doubtless suffice for the solution of the problems investigated in this Note; but would the rules thus determined give the necessary degree of accuracy? I prefer to treat these questions in a manner which is unquestionably less elegant, but in one which leaves no room for doubt as to the results.

SIMILITUDE IN STEAM ENGINES.

When the plans of engines working at a fixed pressure have been drawn to a certain scale, can they be applied on a different scale and give the same assurance of good working?

Let us first examine the statical strains, assuming that the weight of the parts may be neglected in comparison with the strains produced by the steam.

Friction.—The pressures on the moving and rubbing surfaces varying as the square of the linear dimensions, the load per unit of surface will not change.

Resistance of Parts.—For the same reason, the parts which are in compression and tension support the same load per unit of section, whatever the scale may be.

For those which are subjected to bending, the load R per unit of section is:

$$R = \frac{PLd}{I},$$

in which

P is the load,

L the length of the arm,

d the distance of the fiber furthest from the neutral axis,

I the moment of inertia.

P is proportional to the square of the linear dimensions, L and d to the first power only, and I to the fourth; R is therefore independent of the scale.

The same is true of the strains of torsion.

For cylindrical shapes subjected to uniform internal or external pressures, the material is equally loaded when the thickness varies with the linear dimensions. This rule may also be extended to volumes of any form.

So also, in all that concerns statical loads, the material is subject to identical strains per unit of section, and, if the strength is independent of the dimensions, the conditions of safety remain the same, whatever the scale.

Let us now consider the strains due to the inertia of the moving parts. It is hardly necessary to state that what follows does not apply to slow running engines, such as those for paddle wheels.

As these strains do not vary with the scale, it is evident that, like the statical ones, they should be proportional to the square of the linear dimensions.

It will suffice to consider the principal inertia strains, such as those of the pistons and their accessories.

Let F = strain in pounds due to inertia,

P = weight in pounds of the moving parts,

C = stroke of pistons in feet,

N = number of revolutions per second,

φ = angle which the crank makes with the axis of the cylinder produced,

m = ratio of connecting rod to crank.

Then

$$F = -6.59\, PCn^2 \left(\cos.\, \varphi + \frac{1}{m} \cos.\, 2\, \varphi \right).$$

P varying as the cube, and C as the first power of the linear dimensions, in order that F may be proportional to the square, the number of revolutions n must be inversely proportional to those dimensions.

Other things remaining the same, the ordinary rule on which we compute the value of the inertia on the basis of the linear velocity of the piston is therefore correct.

Strains set up in the vessel when in motion due to the inertia of the fixed and moving masses of the engine.—From this point of view, the law of similitude is faulty, and account must be taken of the rapidity and the amplitude of the vibrations of the parts whose resistance is brought into play.

Here one remark is necessary. Some powerful engines set up in very light hulls can not work satisfactorily without being tied to the hull at several points. Such ties are indispensable, not only on account of the weakness of the bed plate, but also on account of the insufficient bracing of the upper parts of the engine to the foundation. As a matter of fact, the hull adds to the solidity of the engine. Without special arrangements, this plan would be inadmissible in very large hulls, on account of the excessive strains which would be brought upon the engine. The dimensions of engines of this kind could not, therefore, be increased indefinitely.

I have built engines on this plan, but those of my latest torpedo boats, although tied to the hull at several points, are tied there for a different purpose. Although of the same appearance, they present greater stiffness than the hull, whose close tying to the engine reduces the vibration, and if fastened solely upon a sufficiently solid bed-plate, they would work under the same conditions as on board ship. The rules given hereafter are then applicable with the following restrictions:

If we consider that angles of roll of 40 to 45 degrees have been observed, sometimes coincident, as in some foreign battleships, with a relatively short period of oscillation, it seems that it would be well to give vertical engines, tied to the hull only by the bed-plate, a resistance against strains of this kind, such that the engine may be set up in the shop, with the axes of the cylinders horizontal, without strain or appreciable bending. In this position, the load per unit of section, on the foundations, is proportional to the linear dimensions. Therefore, the parts subjected to those particular strains should increase with the size

more quickly than the scale indicates. Furthermore, the weight of the engine increases a little more rapidly than the cube of the linear dimensions.

Passages under pressure.—The sections of the steam passages varying as the square, and the volume as the cube, it is necessary, just as in the case of the inertia strains of the moving parts, that the number of revolutions be inversely proportional to the linear dimensions, in order that the steam passages may be independent of the scale of the drawing.

To sum up, and with the limitations above specified, a *drawing of an engine working at a fixed pressure may be used on any scale, without the load on the material and the rubbing surfaces or the steam passages being modified, provided the number of revolutions be made inversely proportional to the linear dimensions.*

The power of the engines, under the same conditions as regards inertia strains, is then proportional to the square of the linear dimensions, and its weight per I.H.P. to the linear dimensions.

It must be remembered that by engine is meant the cylinders and the parts depending upon them: shafts, bed-plates, connecting rods, etc., but not the condensers and boilers.

I have frequently built engines of different sizes from the same drawings by simply changing the scale and figures. A great saving of time and labor is thus effected. Thus, the engines of the Russian torpedo boats *Revel* and *Sveaborg*, built in 1885, are like those of the *Balny* type, the scale having been increased in the ratio of 1.19; and in 1891 the drawings of the engines of the first class torpedo boats, type 147, were used for building those of the twin screw torpedo boats of the *Lancier* type and of the steamer *Augustin Normand*, the scale having been reduced to .9 in the first case, and to .81 in the second.

Metallurgical industry has made such rapid progress in the manufacture of large pieces of cast and forged steel that I should not hesitate to use these last plans in the construction of much larger engines, as high as 3,000 to 4,000 I.H.P., after having introduced in them the improvements which experience has shown to be useful, and keeping in mind also the special strains due to the motion of the ship at sea.

The following table gives the laws of similitude as applied to engines and hulls, the efficiency being assumed to be constant. It is clear that the similitude of hulls should not hold when the absolute size and speed vary; the expressions containing the speed V, and the displacement D, are, therefore, simply approximate. In fine, the proportions of the screws sometimes make it necessary to modify the number of revolutions. So the table has two columns: in the first, the number of revolutions is supposed to be independent; in the second, it is supposed to have a maximum value—that is, such that the engine remains always in the same condition relative to the moments of inertia. The values of the first column only are of general application.

		Proportional values, the number of revolutions being	
		Independent of the linear dimensions of the engine.	Maximum—that is, such that the ratio of the inertia strains to the statical strains shall be constant.
Power,	F	$n\lambda^3$	λ^2
		$V^3D^{\frac{2}{3}}$	
Linear dimensions of engine,	λ	$\left(\frac{F}{n}\right)^{\frac{1}{3}}$	$F^{\frac{1}{2}}$
		$\frac{VD^{\frac{2}{3}}}{n^{\frac{1}{3}}}$	$V^{\frac{2}{3}}D^{\frac{1}{3}}$
Number of revolutions,	n	$\frac{F}{\lambda^3}$	$\frac{1}{\lambda}$
		$\frac{V^3D^{\frac{2}{3}}}{\lambda^3}$	$\frac{1}{F^{\frac{1}{2}}}$
			$\frac{1}{V^{\frac{2}{3}}D^{\frac{1}{3}}}$
Speed of ship,	V	$\left(\frac{F}{D^{\frac{2}{3}}}\right)^{\frac{1}{3}}$	
		$\lambda\left(\frac{n}{D^{\frac{2}{3}}}\right)^{\frac{1}{3}}$	$\left(\frac{\lambda}{D^{\frac{1}{3}}}\right)^{\frac{2}{3}}$
Displacement,	D	$\left(\frac{F}{V^3}\right)^{\frac{2}{3}}$	
		$\left(\frac{n\lambda^3}{V^3}\right)^{\frac{2}{3}}$	$\left(\frac{\lambda}{V^{\frac{2}{3}}}\right)^3$

	Proportional values, the number of revolutions being	
	Independent of the linear dimensions of the engine.	Maximum—that is, such that the ratio of the inertia strains to the statical strains shall be constant.
Weight of engine proper* without condenser	λ^3	$F^{\frac{3}{4}}$
	$\frac{F}{n}$	
	$\frac{V^3 D^{\frac{3}{4}}}{n}$	$V^{\frac{3}{4}} D$
Weight per I.H.P. of engine proper* without condenser, ...	$\frac{1}{n}$	λ
	$\frac{F}{n^2 \lambda^3}$	$F^{\frac{1}{4}}$
		$V^{\frac{1}{4}} D^{\frac{1}{4}}$
Ratio of linear dimensions of the engine to those of the hull, . . .	$\frac{\lambda}{D^{\frac{1}{4}}}$	$V^{\frac{1}{4}}$
	$\left(\frac{F}{nD}\right)^{\frac{1}{4}}$	$\left(\frac{F}{D^3}\right)^{\frac{1}{4}}$

*On account of the strains due to the motion of the ship, and the reduction of the strength of parts of large dimensions, the absolute weight and the weight per I.H.P. should generally vary a little more rapidly than shown.

This table will be useful in getting out rough drafts when we have satisfactory types of engines and hulls, and as similar as possible to that which we have in view. In fact, it allows of the approximate determination of the principal elements, the others being assumed beforehand.

Suppose, for example, that we want to design a cruiser having a certain power and speed, and that the screws will permit us to use the maximum number of revolutions consistent with the moments of inertia: these are the most favorable conditions as regards weight and complication.

The table shows that the linear scale of the engines is proportional to $F^{\frac{1}{4}}$. The height of the engines is thus known from the types selected, and we have at once the lowest position that can be given to the armored deck to admit vertical engines, and what the dimensions and the height of the engines should be. The table further gives the weight of the engine proper as proportional to $F^{\frac{3}{4}}$, and the displacement proportional to $\left(\frac{F}{V^3}\right)^{\frac{3}{4}}$

If the displacement and speed were given, we would find the scale of the engine proportional to $D^{\frac{1}{3}} V^{\frac{2}{3}}$, the power proportional to $V^3 D^{\frac{2}{3}}$, and the weight of the engine proper to $V^{\frac{2}{3}} D$.

But before definitely deciding anything, it is always well to see whether the number of revolutions is suitable for good proportions of screws.

It would be interesting to calculate the coefficients of the several proportional values in the table for the most recent types, and for those which are considered the best.

The ratio of the linear dimensions of the engine to those of the hull is proportional to the $\frac{2}{3}$ power of the speed, and is independent of the absolute size.

Suppose we have a section of a ship of the cruiser type, in which the engine is shown, and let us assume that it is desired to increase the speed from 20 to 22 knots. Whatever the scale of the drawing, or even the displacement, which might be anywhere from 4,000 to 6,000 tons, the linear dimensions of the engine relative to those of the hull should be increased in the ratio of $\left(\frac{22}{20}\right)^3 = 1.152$.

It can at once be seen whether the solution of the problem is possible without greatly modifying the arrangement of the engines, the protection of the hull, or the form of the bottom.

Moreover, it must not be forgotten that the ratio of the length of the engines and boilers to that of the hull increases in the same proportion; the engines will, therefore, extend further toward the fined portions of the bottom, which will increase the difficulty of getting them in, unless the boiler system is divided into two groups, one forward and the other abaft the engines.

It will be seen from the table that, while the weight of the boilers and condensers varies simply as the power F, that of the engine proper varies at least as $F^{\frac{2}{3}}$. In fact, the former weights should increase a little more rapidly than the power, the difficulties in feeding increasing with a number of boilers; and the second weights a little more rapidly than $F^{\frac{2}{3}}$, because the strength of materials decrease slightly as the size is increased, and because the parts subjected to strains which are the result of the vibration

of the ship should increase more rapidly than the scale. The expression for the weight of the motive apparatus is, therefore, at least of the form :

$$aF + bF^{\frac{4}{3}}.$$

The second term being much larger than the first in large engines, it can be readily seen how much the proportion of weight to power, which we frequently use within wide ranges, varies from the truth.

Thus, in my "Note upon the Variation in the Scantlings of Ships," read here two years ago, I fell into this error. I stated that the weight of the motive apparatus complete, in the equation for the displacement, was, like the power, proportional to $L^2 V^3$, L being the linear dimension of the hull and V the speed.

The true expression should have been of the form :

$$aL^2V^3 + bL^3V^{\frac{4}{3}}.$$

The importance of the weights varying as the cube of the dimensions of the hull being greater, it would have then been found that the advantage of absolute size from a speed point of view is less, and that, on account of the weights of the framing, which should vary partly as the 4th power of the dimensions, the displacement corresponding to the maximum speed is less than I supposed it. The opinion which I expressed upon the futility of indefinitely increasing the dimensions of large steamers in order to increase their speed alone, aside from any commercial consideration, is therefore strengthened.

We have seen that the ratio of the linear dimensions of the engine to those of the hull is simply proportional to $V^{\frac{4}{3}}$; that is, that it is independent of the absolute dimensions, and that it increases rapidly with the speed.

Now, while in large, fast, twin-screw steamers, the engines reach as high as the spar deck, the vertical engines of many cruisers of equal and even greater speed have to be placed below an armored deck, placed at the height of the water line; and even if the engines of the former, like those of the latter, should be lowered as much as the shape of the hull permits, the disproportion in height of the engines would still be excessive. In the

mail steamers, with triple expansion engines, the stroke is sometimes equal to the diameter of the intermediate cylinder; in cruisers, it is always very much less than the diameter of the high pressure. Ought we then to be astonished when engineers, whose commercial engines work for six, eight and even twenty days without stopping, consider themselves fortunate when they have successfully finished a full speed trial of a few hours' duration in a naval vessel?

The stroke is really the most useful linear dimension for good running and for an increase of mechanical duty, and it is almost impossible to give it sufficient value with the system of protection generally adopted, whatever means are employed. One consists in dividing the power between two engines placed in line with each other on the same shaft. Theoretically, the height is reduced in the proportion of $\sqrt{2}$ to 1; but on account of the fining of the hull, and sometimes also on account of the inclination of the shaft, the lengthening of the engines reduces, in a great degree, the apparent advantage. This arrangement carries with it, besides, the great disadvantage of doubling the number of parts, and consequently the chance of accident—an accident to one part involving the stopping of the entire apparatus. The arrangement is in no sense to be compared with two independent engines.

The reduction of the dimensions in the ratio of $\sqrt{2}$ to 1 presupposes that the engine is in the same condition as regards the forces of inertia; that is, that the number of revolutions is increased in the inverse ratio of the dimensions. If, on the contrary, the number of revolutions remains the same as with a single engine, either for the purpose of improving the running, or because the proportions of the screws demand it, the reduction of the dimensions would be only in the ratio of $\sqrt[3]{2}$ to 1.

The other method consists in returning to horizontal or slightly inclined engines, notwithstanding the serious disadvantages which accompany them, and the fact that this arrangement does not give suitable proportions of engines when the speed required is very high.

The engineer to whom we have denied sufficient height, while

waiting for the time when he will be able to use higher pressures than he can now, finds himself fatally led to adopt a stroke that is too short, too great a number of revolutions, or to double the number of separate engines on the same shaft: three factors which increase the chances of wear and accident to engines. Space is as necessary for horse power as for horses themselves; it is as indispensable for the endurance of the personnel as for the incessant surveillance of the machinery. But the contest for speed is ever on. The continual reduction of weight of boilers, and progress in the economical return from engines, lead to the reduction of weights, which all navies are tempted to make use of in order to increase speed. The protective deck near the water line, which yesterday was an annoyance, to-morrow will become an absolutely insurmountable obstacle. Evidently we must get rid of it; it can be done without regret.

Is it well to always economize in weight in order to get an increase of maximum speed? At the present speeds, if we gave vertical engines the proportions that they should have, they would come way above the water line. The armor, especially that on the sides, is indispensable, not only for the protection of the motive machinery, but, what is much more important, to insure the floatability and the stability of the ship, and would absorb the greater part of the saving in weight; and the importance of this armor would increase very rapidly with the dimensions of the engines, that is to say, with the maximum speed. If any weight is left at the disposition of the architect, should it not rather be utilized in increasing not the maximum speed, but that which the ship can maintain in heavy weather?

An increased weight of hull would undoubtedly result therefrom because this result could not be obtained solely by forming the stern lines properly, immersing the screws so as to keep them always submerged, and in so arranging the top sides that no dangerous mass of water can be retained there. The scantlings of the frame and deck erections must also be generally increased.

The reduction of the fineness, a consequence of the reduction of maximum speed, would furnish an augment of displacement of quite 3 per cent. per knot. The ship could then get through

any sea without distress, just as the mail steamers, which the war ship is destined to capture whatever the condition of the sea may be, now do, and the state of the sea does not increase their time of voyage more than a few hours during the worst weather.

If the power is not increased, the arrangement of the engines should be less complicated, and it would then be possible to give them proportions consistent with good working.

ON THE SIMILITUDE IN THE TRIALS OF MODELS AND OF SHIPS.

Reech first, and Froude afterwards, showed that, as regards the direct resistance, and more especially as regards the formation of waves, the "corresponding speeds" of the model and the similar ship are proportional to the square root of the linear dimensions."*

It is interesting to see whether this law holds as regards lifting, or the reduction of displacement which seems to accompany very high speeds, and also for the phenomena attending the rupture of the column of water put in motion by the propellers, upon which I addressed the Association a year ago.

Lifting.—In every ship in motion, there is a vertical component of the direct resistance, which tends to raise the bow and to lower the stern. The forward component increases indefinitely with the speed, while the after one tends towards a finite limit, which is reached when a complete vacuum is obtained in this part. At very high speed, the former is greater and causes lifting.

The richochet of projectiles is the most familiar demonstration of this phenomenon.

Does this effect manifest itself at high speed in light vessels, such as torpedo boats? It is probable, although the increased efficiency may be due to other causes.

If it does manifest itself, it is clear that the like points of the phenomonon are those in which the ratio of the direct resistance to the displacement is the same in the model and in the ship, for the direction of the resisting forces cannot be different in the model from that in the ship. The following table shows that it

*In 1870 I extended this law to variations in coefficient of fineness.

is so when the speeds are proportional to the square root of the linear dimensions:

	Model.	*Ship.*
Linear dimension,	1	Λ
Resisting surface,	1	Λ^2
Displacement,	1	Λ^3
Speed,	1	$\Lambda^{\frac{1}{2}}$
Direct resistance,	1	$\Lambda^2(\Lambda^{\frac{1}{2}})^2=\Lambda^3$
Ratio of direct resistance to the displacement,	1	1

Therefore, *if the speeds of the model and of the ship are increased indefinitely*, their displacement will be reduced by the same fraction at speeds which are proportional to the square root of the linear dimensions.

In the trials made at Brest, the greatest speeds obtained scarcely exceed the half of that at which the phenomenon would have clearly manifested itself, or, what amounts to the same thing, the linear dimensions of the models are three or four times too great.

RUPTURE OF THE COLUMN OF WATER SET IN MOTION BY THE PROPELLERS.

Let us begin by establishing the fact that the slip appropriate to the speed is the same when the latter is proportional to the square root of the linear dimensions.

	Model.	*Ship.*
Linear dimensions,	1	Λ
Propulsive surface in square feet,	A	$\Lambda^2 A$
Resisting surface in square feet,	S	$\Lambda^2 S$
Speed in feet per second,	V	$\Lambda^{\frac{1}{2}} V$
Speed of water acted upon in feet per second,	V^1	$\Lambda^{\frac{1}{2}} V^1$

If we designate by R the resistance in pounds per square foot of resisting surface, which is the same in each case, the resistance of the model in pounds, assuming no "cavitation," is

$$RSV^2=\frac{43{\cdot}3\ AV^1}{g}(V^1-V),$$

and that of the ship

$$R\Lambda^2 S\Lambda V^2=\frac{43{\cdot}3\Lambda^2 A\Lambda^{\frac{1}{2}}V^1}{g}(\Lambda^{\frac{1}{2}}V^1-\Lambda^{\frac{1}{2}}V).$$

The two equations are identical, which proves the correctness of the proposition. Therefore, the speed of the slip of the water is, like the speed of the ship, proportional to the square root of the linear dimensions.

Now, the rupture of the column of water is produced when the speed of replacement, which varies as the square root of the height of the point considered below the water line (linear dimension), becomes less than the speed of the ship.

Therefore, for every similar point of the screws, the rupture takes place in the model and in the ship at speeds which are proportional to the square root of the linear dimensions.

The theory of similitude is thus extended to the two phenomena of lifting and of rupture of the column of water propelled. The augmentation of resistance due to the sucking of the screws, and the longitudinal inclination due to the speed, also follow the same law.

It is, therefore, probable that, in general, it is safe to admit complete similitude, except as regards friction.

The propellers have such an influence upon the resistance, and their dynamic effects are so important for efficiency, that, if a more complete study should confirm this opinion, it would probably be necessary to replace traction trials of models by propulsive trials. The latter would give results that could be immediately applied, which is far from the case now.

While waiting until we can transmit an exactly measurable electric power, capable men, of whom there are many, will certainly construct sufficiently powerful and light mechanical motors which will give a series of couples of uniform constant rotation. The course need be only long enough for the determination of the speed of the model after it has obtained a uniform rate. An initial impulse would greatly reduce the time necessary for reaching a uniform speed, and consequently the length of the course. We might, perhaps, be able thus to more easily elucidate the phenomena which are present at extreme speeds.

The expense of such trials would probably be less than those of towing, as a permanent outfit would be unnecessary.

THE DETERMINATION OF THE DRYNESS OF STEAM.

By Professor W. Cawthorne Unwin, F. R. S.

[Paper read before the Institution of Mechanical Engineers.]

Every engineer who has to do with the manufacture or use of steam machinery will know that a large amount of experimental research has been carried out during the last ten or fifteen years, on the action of steam in the engine, and on the causes of waste of energy in the production and application of steam power. The testing of boilers and engines with a view to determine their comparative economy has come to be a matter of considerable practical and commercial importance. Not only are boilers and engines contracted for under stringent guarantees as to the amount of coal and steam they will use, but the results of trials are published in a kind of open competition; and there is a tendency to record-breaking in engine tests, as well as in trials of a more sporting kind. For commercial reasons, therefore, as well as for scientific reasons, it is important that the observations taken in an engine or boiler trial should be complete and accurate, and that the instruments used should be of the most satisfactory and trustworthy character.

In such trials there has been one important observation which has hitherto been difficult, and which has been made by methods of doubtful trustworthiness. The dryness of the steam is a quantity which, in either a boiler trial or an engine trial, it is important to determine. If a boiler produces wet steam, it is credited with greater evaporative efficiency than it deserves, if the amount of unevaporated water is not ascertained. If an engine is supplied with wet steam, its thermal efficiency will be diminished, and it will be undeservedly discredited unless the quality of the steam is known. It has long been understood that it is desirable in both

boiler trials and engine trials to determine the quality of the steam; but the methods for measuring the amount of moisture in steam were troublesome to carry out, and when used they gave results which were more or less discordant and doubtful. It occurred to the author to suggest at the Edinburgh meeting of the British Association that a committee should be constituted to examine the various methods of determining the dryness of steam. A committee was formed with Sir Frederick Bramwell, Bart., as chairman; and a report was presented at last year's Oxford meeting. The present paper contains some account of the methods described in that report, and of the conclusions arrived at in trials of different methods.

The earliest attempts to determine the amount of moisture in steam, of which records have been found, were made during some boiler trials carried out by a committee of the Société Industrielle of Mulhouse, in 1859. This committee tried three different methods—a method of separation, a condensing method suggested by Hirn, and a chemical method. In these early trials the condensing method only, in which the total heat of a sample of the steam was measured, appeared to give satisfactory results. But although the committee did not place full reliance on any of their methods, these have all been used by various experimenters down to the present time.

Origin of the Water Suspended or Entrained in Steam.—The water found in a sample of steam may have come there in one of three ways.

1. Water projected into the steam space during ebullition may be carried forward in the current of steam. The extent to which wetness is thus produced depends on the activity of the ebullition, the area of the water surface, the volume of the steam space, the position of the steam valve, the density of the steam, and, probably more than anything else, on the quality of the water and its liability to produce foam. Mr. Thornycroft has made some instructive observations on the priming produced when water foams. He constructed a boiler with glass ends, through which the process of boiling could be seen. As the result of observations on this boiler he states that waters which

cause priming produce foam on boiling. Water which is very bad produces bubbles so durable as to remain a considerable time without breaking; and by them the steam space of a boiler may be entirely filled. So soon as this takes place, instead of simply steam leaving the boiler, the discharge consists of foam, which becomes broken up in its rapid passage through the steam pipe. With pure water, steam retains no film of liquid long enough to be seen.

2. Water may be produced in steam from the expansions to which it is subjected. Fluctuations of pressure arise from the intermittent demand for steam, and from the steam passing from places of higher to places of lower pressure. But it is difficult to believe that any great amount of wetness arises in this way in ordinary cases.

3. The steam in the steam space of the boiler, and when flowing through the steam pipes, loses heat by radiation from the boiler roof and the surfaces of the pipes, and there is consequently a condensation of part of the steam. Probably in some cases considerable wetness is produced in this way. In the case of any individual steam plant, the absolute amount of moisture produced by radiation in a given time will be constant and independent of the demand for steam. The wetness of the steam, therefore, so far as it is due to this cause, will increase as the demand for steam diminishes.

METHODS OF DETERMINING THE WETNESS OF STEAM.

1. *Weighing Method.*—The density of saturated or very approximately saturated steam, of the quality of that in Regnault's total heat experiments, can be directly determined on thermodynamic principles, and the result is confirmed by Fairbairn's experiments. Hence, if a known volume of steam is weighed, any excess of weight above that of a corresponding volume of dry saturated steam must be due to the water present. A method of direct weighing has been proposed by Guzzi ("Revue Industrielle," 1878, page 102), and by Knight ("Journal of the Franklin Institute," 1877, page 358). The method is obviously one of excessive difficulty.

2. *Separating Method.*—It is common to separate in a steam separator a great part of the water entrained in steam. In ordinary steam separators, however, the volume passing through is very large compared with the capacity of the separator, and the action is no doubt imperfect. In a separator dealing with only a small fraction of the steam, more perfect separation might be expected. Mr. G. A. Barrus, using a small separator in connection with a superheating calorimeter, noticed that nearly all the entrained water was trapped, and that the steam passing into the superheating vessel was nearly dry. More recently, Professor Carpenter, of Cornell University, has introduced a form of separating calorimeter of small size, which appears to give highly consistent and trustworthy results, and can be used with great facility. It consists of a vessel A, Fig. 1, about 12 inches high by 5 inches in diameter, consisting of an inner chamber and a jacket. The steam from the steam pipe S passes first into the inner chamber, where the moisture is separated, and then into the outer chamber. The separating chamber is therefore perfectly protected from radiation. As the water accumulates in the inner chamber its level is shown by a gauge glass G, and the amount in hundredths of a pound can be read off on a scale. A very small orifice at the bottom of the outer chamber regulates the amount of steam discharged. The escaping steam passes through a flexible tube to a simple form of condenser C. The increase of weight in any given time in the condenser is noted, and the amount accumulated in the same time in the separator. If x is the dryness fraction of the steam, w the weight of water caught in the separator, and W the weight of steam condensed, then $x = W \div (W + w)$. There is a gauge glass and scale on the condenser, graduated to read pounds and tenths at a temperature of 110 degrees Fahrenheit. But as the variation of volume in the condenser with temperature affects the readings considerably, it is best to place the condenser on a platform weighing machine. Professor Carpenter states that the dryness of the steam after passing the separator was tested in the laboratories at Sibley College by several observers, and with steam carrying from $\frac{1}{4}$ per cent. to 60 per cent. of moisture. In every case the separation

of the water from the steam was complete and perfect. Other tests have been made with moderately dry steam, using the throttling and separating calorimeters simultaneously, and the results were practically identical. The instrument is very simple to use, and requires no pressure gauge or thermometers.

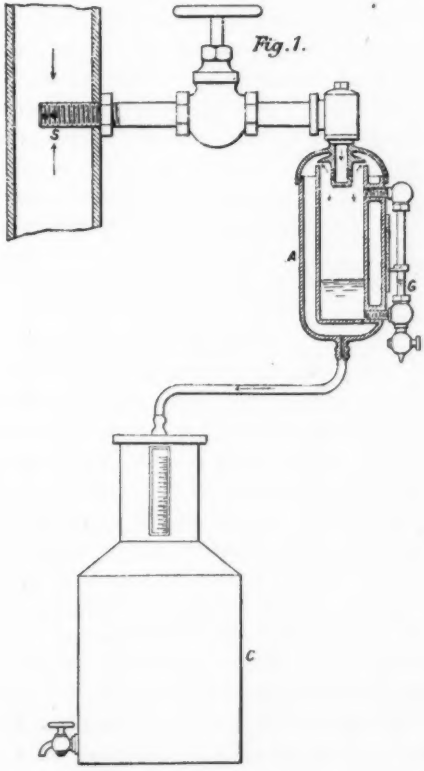
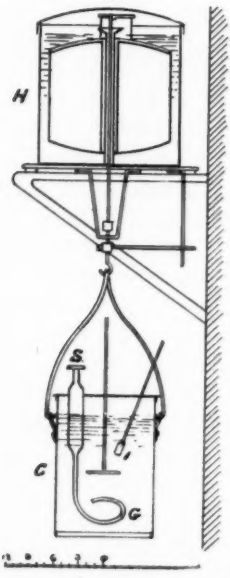


Fig. 1.

3. *Condensing Method.*—Suppose a known weight of the steam to be condensed, and its total heat to be determined by the rise of temperature of the condensing water. By comparing the total heat per pound of a sample of steam with that of a pound of dry

saturated steam according to Regnault's tables, the amount of moisture in the steam can be determined. This method was first suggested by Hirn, and the apparatus which he designed for use in the Mulhouse Boiler Trials of 1859 is, perhaps, the most convenient form of apparatus for determinations by this method. It consists of an iron vessel *C*, Fig. 2, about 1 foot in diameter, furnished with a loose cover; this forms the condenser. A small pipe and cock in the steam pipe deliver steam through a small orifice near the steam pipe into the pipe *S*, through which it passes into the condensing water. An agitator *G* and a sensitive thermometer *I* are provided in the condenser. For weighing the amount of steam condensed, the whole condenser is suspended from a hydrostat *H*, which permits extremely accurate determination of any change of weight. The hydrostat is balanced by weights till the pointer is at a fixed mark before and after condensing the steam.

Fig. 2



Let x be the dryness fraction of the steam, w the increase of weight of the condenser during the test. Then the condenser has received xw pounds of dry steam, and $(1-x)\,w$ pounds of water at the steam temperature. Let W be the weight of water initially in the condenser, plus the equivalent of the condenser itself reckoned as water. Let t be the temperature of the steam, and i and f the initial and final temperatures of the condenser water. Then using the ordinary approximate equations,

$$xw\,(1{,}116-0.71\,t)+w\,(t-f)=W\,(f-i);$$

$$\text{whence } x=\frac{W\,(f-i)-w\,(t-f)}{w\,(1{,}116-0.71\,t)}.$$

To arrive at satisfactory results, the temperatures must be read

to $\frac{1}{10}$ degree Fahrenheit at least, and the weight of steam condensed must be very accurately determined. It is desirable that the initial temperature i of the condenser water should be about as much below the temperature of the place where the test is made as the final temperature f is above it. A correction for radiation during the time that the agitator is used before the final temperature f is read can be made thus: Suppose the agitator is used for two minutes, and the temperature is then f_1, and that two minutes later a temperature f_2 is observed. Then the true value of the final temperature f at the moment when condensing ceased is $f = 2f_1 - f_2$. In the Mulhouse tests the initial and final temperatures were about 60 degrees and 110 degrees Fahrenheit, and the steam generally showed from $2\frac{1}{2}$ to 5 per cent. of moisture, but in particular cases as much as 12 to 16 per cent. of moisture was observed.

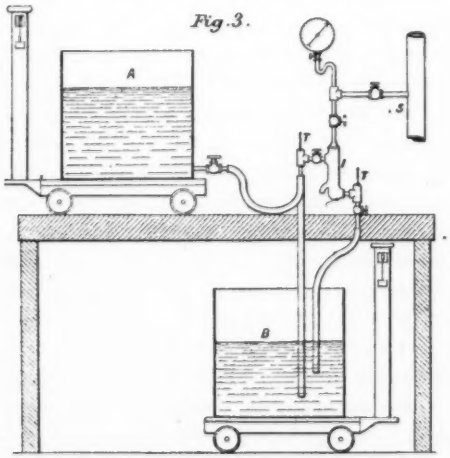
The condensing or total-heat method of ascertaining the dryness of steam has been used by many observers during the last thirty years. An apparatus for making this test is known in America as the barrel calorimeter. An ordinary oil barrel is used, fitted with an outlet valve. The barrel is filled with cold water, and steam is led into it by a pipe and condensed. The barrel is placed on a weighing machine, and the weight and temperature before and after condensing are noted. The proceeding is a rough one, and has given anomalous results in practice. Mr. Willans used this method in his trials of a non-condensing steam engine. He sought to evade the known difficulties of the method by carrying it out on a very large scale. The condensing tank, weighing with water about three tons, was placed on a platform weighing machine. This was balanced by the sliding weight of the weighing machine. Then a standard hundredweight was placed on the platform, and the machine lever again put in balance. The hundredweight was then removed, the temperature noted, and steam condensed till the added weight of condensed steam just lifted the lever, making an electric signal. Then the water was stirred, and the temperature again noted. The thermometers were very sensitive, and were accurately compared with standard thermometers. A small correction was made for radiation.

The value of the dryness fraction found on different days was 0.9986, 0.9638, 0.9949, 0.9646, 0.9976, 0.9893, 1.0072, 1.0048, 0.9987. The mean of all was 0.9911, showing an average of about 1 per cent. of moisture in the steam. Probably this test was never more carefully carried out; but the results are not so accordant as could be wished. Two of the results indicate superheating, which under the circumstances was probably impossible.

The condensing or total-heat method is strictly accurate in principle, but it is difficult to carry out satisfactorily. It is not easy to make the proper allowances or corrections for the heat absorbed by the condensing vessel itself, and for radiation and loss by evaporation from the condenser. The thermometers must be very sensitive; and very sensitive thermometers are difficult to use.

4. *Continuous Condensing Methods.*—The difficulties of the ordinary condensing method were so obvious that Mr. Barrus, Mr. Hoadley and other observers were led to propose methods of continuous condensation. Steam and cold water being both supplied at a constant rate, the condenser requires a steady temperature, which can be very accurately observed. The steam may be condensed either in the condensing water or in a surface condenser. Fig. 3 shows a continuous injection condenser.

Fig. 3.



Steam passes from the steam pipe S to a small injector I. The condensing water is drawn from the tank A, and the mixed water and condensed steam are discharged into the tank B. The two tanks are placed on platform weighing machines. Thermometers T, T give the temperatures of the condensing water and of the mixture of condensed steam and condensing water. The difference of the total weight in the two tanks after any interval of time is the steam condensed in that time. Fig. 4 shows a continuous surface condenser, consisting merely of a small pipe A in a vessel, through which flows a steady stream of condensing water. As the condensing surface is constant, the rate of condensation is constant, and the rise of temperature of the condensing water is constant. The condensed steam is drawn off steadily from the condenser for weighing, and simultaneously a series of readings are taken of the water entering and leaving the condenser. The continuous condensing method seems likely to be more accurate than the ordinary method; but it involves more elaborate arrangements, and it does not seem to have been much used in practical trials.

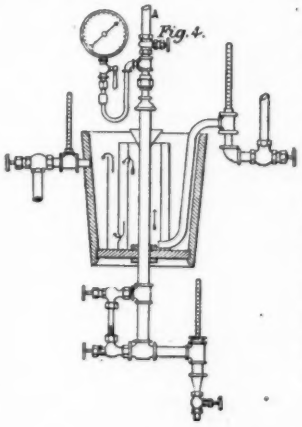


Fig. 4.

5. *Superheating Method.*—About 1890 Mr. G. H. Barrus devised a calorimeter, in which the steam to be tested passed through an inner chamber jacketed by superheated steam. The sample of steam to be tested was thus dried and superheated at the expense of heat borrowed from the jacket. To avoid measuring the steam, an attempt was made to secure that equal weights of steam passed through the inner chamber and through the jacket. In that case the wetness of the steam can be calculated from observation of the temperatures only. The method is accurate in principle, but appears to be difficult to carry out satisfactorily.

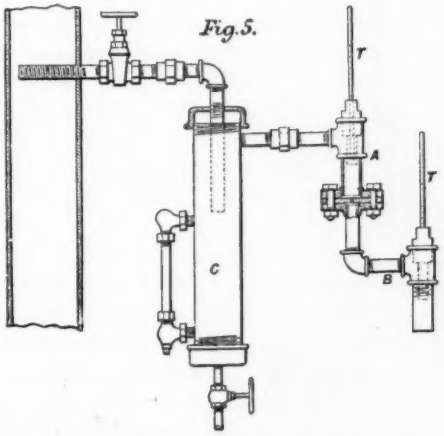
6. *Wiredrawing Method.*—A method in which the steam to be tested is dried and slightly superheated by wire drawing was proposed by Professor Peabody. Mr. Barrus and others have devised modified forms of the apparatus. Fig. 5 shows Mr. Barrus's arrangement. The steam passes from a chamber A to a chamber B through a very small aperture, about $\frac{1}{16}$ inch in diameter. The full steam pressure is in A, and the pressure in B differs little from atmospheric pressure. Thermometers T,T give the temperatures in the chambers, which are protected from radiation by a thick coating of asbestos and felt. The steam is allowed to flow through the apparatus for 20 minutes or more, when the temperatures become nearly steady. Let t_1, t_2 be the temperatures of the steam before and after wiredrawing, t_3 the temperature of saturated steam corresponding with the pressure in the second chamber B. Then the steam in B has been superheated $t_2 - t_3$ degrees by wire drawing. Let h_1, L_1 be the total heat and latent heat of steam at t_1, and h_3, L_3 corresponding quantities for steam at t_3 Taking as usual the specific heat of water as unity, and that of steam as 0.48, the dryness fraction $x = \frac{t_3 - t_1 + L_3 + 0.48\,(t_2 - t_3)}{L_1}$. No weighing is required, and temperatures only have to be observed. The observations can be continued as long as desired, so as to obtain a mean value for the dryness fraction x from a considerable quantity of steam. If the steam is very wet, the temperature in B falls to about 212 degrees, showing that wiredrawing to atmospheric pressure is insufficient to dry the steam. Practically, the instrument cannot be used if the wetness exceeds the values given in the following table, the pressures being in pounds per square inch, and the atmospheric pressure being assumed at 14.7 pounds.

Initial Pressure (Absolute).	Initial Pressure (Gauge).	Initial Temperature.	Initial Wetness.
		Deg. Fahr.	*Per Cent.*
29.9	15 2	250	0.80
67 2	52.5	300	2.44
135.1	123.4	350	4.21
247.7	233.0	400	6.13

Two conditions are necessary for accuracy in using this method. The chamber B must be large enough for the eddies to die out before the steam leaves the chamber. Radiation must be so far prevented that the steam in the chamber is not sensibly cooled. As a large quantity of heat is passing through the chamber, it appears that, when reasonable precautions in clothing the apparatus are taken, the loss by radiation is so small a fraction that it produces no important effect.

7. *Combined Separating and Wiredrawing Methods.* — To extend the usefulness of the wiredrawing method, Mr. Barrus added a separator C, Fig. 5. The steam in passing through the

Fig. 5.



separator leaves most of its moisture there; and the remainder is measured in the wiredrawing part of the instrument. When thus arranged the use of the instrument is much more troublesome, as the amount of steam passing through in a given time must be observed, and the amount of moisture collected in the same time must be weighed. A condenser must therefore be used, at least occasionally, in order to determine the discharge per minute by the apparatus at different pressures. A formula can be found, connecting the discharge per minute with the fall of pressure in the chambers A and B; but it is not clear that the

calculation of the steam used from the formula is quite accurate enough for the purpose; and using the condenser in every test is troublesome.

Fig. 6 shows a modification of Mr. Barrus' apparatus, which is manufactured by the Globe Engineering Company. A gauge glass is added to the separator, for facilitating the observation of the quantity of steam trapped. At the bottom of the separator is a cock, by which the water can be drawn off, or the level in the gauge glass regulated. The Globe calorimeter is well arranged and neatly made; but a condenser should be supplied with it, and be capable of easy attachment to the discharge pipe. The separating and wiredrawing parts in this instrument cannot be used independently.

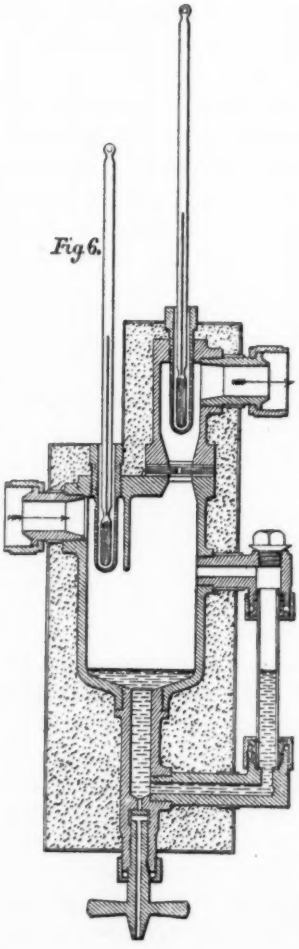


Fig 6.

8. *Second Superheating Method.*—Mr. W. R. Cummins has suggested another method. A vessel is filled with the steam to be tested, and then heated by a jacket. As it is heated, the rise of pressure in the inner vessel is observed, the volume being constant. So long as the steam is moist, the pressure will rise with the temperature according to the law for saturated steam. The moment all the moisture is evaporated, the rate of rise of pressure with temperature will become much slower. Both inner chamber and jacket are first blown

through with steam from the steam pipe. Next both are closed, and heat is supplied to the jacket. During this process the temperature and pressure in the inner chamber are observed at short intervals of time; and the temperature t_2 is found, at which, from the change in the rate of increase of pressure, the steam has just become dry. Let t_1 be the initial temperature of the steam, and v_1, v_2 the specific volumes of saturated steam corresponding to the temperature t_1, t_2. Then the initial dryness fraction x of the steam is $x = v_2 \div v_1$. The method is correct in principle, and there does not seem to be any insuperable difficulty in using it; but it does not appear that it has been tried.

9. *Chemical Methods*—Let a soluble salt be added to the boiler water, so as to form a solution of known strength at any given time; and let the boiler be afterwards fed with pure water. If the steam leaving the boiler is dry, saturated steam, the boiler solution should remain of constant strength. But if there is priming, part of the boiler water will be removed with its percentage of salt, and the solution in the boiler will diminish in strength. This method has been often used from the date of the Mulhouse experiments in 1859 to the present time. There are three variations of the mode of procedure.

(*a*.) The decrease of saltness of the boiler water in a given time can be determined by taking samples from a gauge cock and analyzing them. The sample should be drawn off through a worm cooled so that the water is below 212 degrees at the point of discharge; otherwise there will be a loss by evaporation, which will alter the saltness of the sample. Care should be taken that the level in the boiler is the same when the initial and final samples are taken.

(*b*.) A sample of the steam may be condensed in a small surface condenser, and the amount of salt present in the condensed steam can be determined. Numerous samples should be taken during a trial, so as to obtain an average value of the dryness fraction of the steam.

(*c*.) Mr. Escher, of Zurich, has suggested another mode. The boiler is fed with water containing a constant percentage of sol-

uble salt. During working the solution in the boiler steadily concentrates. If the boiler primes, the concentration will reach a fixed limit, when the amount of salt in the concentrated solution removed by the priming water is as great as the amount of salt introduced in the same time by the weaker feed water. If the limit of concentration and the quantity of feed supplied are known, the dryness of the steam can be exactly determined, at least so far as the moisture in it is due directly to priming. Mr. Escher has shown how the dryness can be approximately determined from the concentration in a given time, when the limit is not reached. This appears to be accurate in principle, as a means of determining the amount of mechanical priming; but no record has been found of its having been tried.

The author has examined the results obtained by the salt methods (*a*) and (*b*) in a number of boiler trials, and he believes that these justify the following general conclusions: 1. According to the salt test there is usually less than $\frac{1}{5}$ per cent. of moisture in the steam produced; whereas in tests of the same steam by other methods a considerably greater amount of moisture is indicated. No doubt this arises partly, perhaps principally, from the fact that the salt test can show only that part of the moisture in the steam which is due to mechanical priming. It may be inferred that in ordinary cases a sensible proportion of the moisture in steam is due, not to mechanical priming, but to condensation in consequence of radiation occurring after the steam is formed. 2. The results obtained by salt tests during any one boiler trial are not closely accordant, and results by method (*a*) do not well agree with results by method (*b*). This throws some doubt on the accuracy of the methods. 3. In cases where there was obviously a good deal of mechanical priming, the wetness shown by salt tests in successive samples of the steam is extremely variable. In a trial by Dr. Bunte, for instance, tests by method (*b*) showed from no wetness up to 13 per cent. of moisture, the mean being $3\frac{1}{4}$ per cent. At the same time method (*a*) gave 1.7 per cent. of moisture. It is obvious that in method (*a*) there must be great difficulty in securing a uniform distribution of salt in the boiler, and if this is not obtained it is impossible

to get average samples of the boiler water. The fact that the feed is supplied at one definite place tends to prevent a uniform distribution of saltness. In method (*b*) it would seem that the amount of steam which can conveniently be condensed is not large enough to be an average sample of the steam. A subordinate question is this. Some engineers have thought that the salt method was most suitable for boiler trials, because it gives directly the mechanical priming; while other methods were more suitable for engine trials, where a knowledge of the absolute dryness of the steam was required. It seems doubtful if the salt test does give the mechanical priming accurately; but also the view seems to be founded on a misapprehension. In determining the evaporative efficiency of a boiler it is necessary to know how much of the feed leaves the boiler as steam and how much as water. The total heat utilized depends on the dryness of the steam leaving the boiler. It does not matter how moisture originates in the steam, provided it is there. Moisture produced by radiation from the boiler roof is as much a deduction from the efficiency of the boiler as moisture projected into the steam mechanically. It is important in a boiler trial that the steam tested should be taken very near the boiler, and not from a steam pipe in which heat may have been lost by radiation. On the other hand, in an engine trial it is desirable that the sample of steam should be taken very near the engine.

PRACTICAL CONCLUSIONS AS TO THE MOST CONVENIENT AND ACCURATE METHODS.

The chemical method, the author thinks, may be put aside as both inconvenient and untrustworthy, except, perhaps, in the single case of a boiler subject to marked priming action. In that case the use of the salt test according to method (*a*) might be useful, because, virtually, it integrates the loss of boiler water by mechanical priming over any desired length of trial, and because, where the wetness of the steam is large, the defects of the method, especially the difficulty of securing average samples of the boiler water, are proportionately less serious.

Of the other methods, the condensing or total-heat method,

accurate as it is in principle, must be rejected on the ground of the difficulty of making the observations with the necessary accuracy.

Putting aside some other methods which are almost untried, and which hardly promise to be convenient, there remain two methods only of determining the dryness of steam, which seem to fulfil the necessary conditions. These two methods are the wiredrawing method and the separating method. For either of them very convenient apparatus has been arranged, and the observations to be taken are simple.

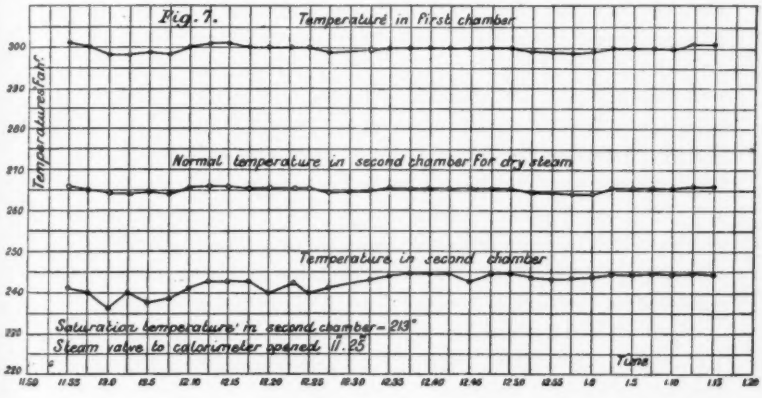
Tests of the Wiredrawing Calorimeter.—This apparatus is so easily used, and gives indications of such great delicacy, that it seemed desirable to get a direct test of its trustworthiness by using a sample of steam, about the quality of which there could be no doubt. Especially was it desirable to ascertain whether the small loss of heat by radiation caused any appreciable loss of temperature in the steam in the second or superheating chamber. It occurred to the author to superheat a sample of steam on its way to the wiredrawing calorimeter, and to observe whether the change of temperature in the instrument corresponded with that which should occur according to calculation from the quantities of heat concerned. A small superheater was constructed, heated with gas jets. The steam being passed through this, there was superheated steam in both chambers of the instrument.

Let p_1 be the pressure in the first chamber *A*, Fig. 5, before wire drawing; t_1, the corresponding temperature of saturated steam; t'_1, the actual temperature of the steam. Then the steam entered the instrument with $t'_1 - t_1$ degrees of superheat. Let p_2, t_2, t'_2 be the corresponding quantities for the second chamber *B*. Then, after wire drawing, the steam has $t'_2 - t_2$ degrees of superheat. Since the total heat per pound is the same for the steam passing through both chambers, if radiation is neglected, and if the frictional eddies are destroyed,

$$1{,}082 + 0.305\, t_1 + 0.48\,(t'_1 - t_1) = 1{,}082 + 0.305\, t_2 + 0.48\,(t'_2 - t_2).$$
$$\therefore\ t'_1 - t'_2 = 0.3646\,(t_1 - t_2).$$

The quantity $t_1 - t_2$ can be found from the observed gauge pressures, and the observed $t'_1 - t'_2$ can then be compared with

the value here calculated. Several tests were made in the laboratory of the Central Technical College, in all of which the observed and calculated values of $t'_1 - t'_2$ agreed very closely. This shows that the effect of radiation in reducing the temperature in the second chamber B is very small. Probably this is due to the fact that the heat lost by radiation is small, compared with the amount of heat passing through the apparatus. In Fig. 7 are shown curves drawn for one of these tests. The steam was taken from a steam pipe supplying an engine from a Babcock and Wilcox boiler. The engine was running, and the steam pressure was taken by a Schaeffer and Budenburg gauge,



and can, therefore, be approximately accurate only. The mean superheat in the first chamber A was 24 degrees; that in the second chamber B was 80.1 degrees. The mean observed difference of temperature in the first and second chambers was 30.78 degrees, and the value calculated from the above formula was 31.45 degrees, showing a loss, if the pressure observations are assumed to be accurate, of only 0.67 degrees, possibly due to radiation. But 1 per cent. of the moisture in the steam would have caused a lowering of temperature in the second chamber amounting to about 20 degrees. Even, therefore, when used without any correction for radiation, it would appear that the wiredraw-

ing calorimeter gives results accurate within about 0.1 per cent. of moisture, provided that a fair sample of steam is obtained. Hence it appears that practically radiation may be neglected, and that for steam initially wet the equation in section 6 may be used.

Test of the Efficiency of a Separator.—The Barrus calorimeter, which the author obtained from America, has a separator *C*, Fig. 5, which can be used in conjunction with the wiredrawing part of the apparatus. This separator does not seem to be quite so well arranged as that of the Carpenter colorimeter; but it occurred to the author that it might be useful to observe how far the separator trapped the moisture of the steam, and what fraction passed on to the wiredrawing aperture. In three tests made at the Central Technical College the separator trapped from 4.4 to 9 per cent. of moisture, and the steam passing through the wiredrawing aperture showed only from 0.4 to 0.2 per cent. of moisture. It would appear, therefore, that even with very moist steam the separator alone will give results not erring by 0.5 per cent. of moisture. In the Carpenter calorimeter, Fig. 1, the current of steam is slower, and the separator is steam-jacketed, and the author thinks its action is more perfect. According to Professor Carpenter's results already mentioned, the action of the separator is practically so perfect that it traps all the moisture.

Tests of a Wiredrawing and Separating Calorimeter Side by Side.—The author has tried the two calorimeters side by side, the steam current dividing at a T-piece to flow to them both alike. In these tests the results by the two methods agreed fairly well, but the wiredrawing calorimeter showed rather more moisture than the separating calorimeter. The former used much more steam than the latter, and the author thinks it possible that a rather larger percentage of moisture was carried away by the stronger current at the T-piece. An incidental result in making these tests was interesting. The steam was taken from the engine steam pipe to the calorimeters through a branch pipe about 30 feet long, which was covered with Keenan's composition. When the separating calorimeter was used alone, the steam showed 13.82 per cent. of moisture. When both calori-

meters were in use together, the steam showed only 3.93 per cent. of moisture. The difference appears to be due to radiation from the 30-foot length of steam pipe; with the very small current required for the separating calorimeter, the condensation due to radiation produced a considerable effect; with the much faster current required for the two calorimeters used together, the effect of radiation was much less. So must it be in the working of steam stations: When the engines are working at full power, the effect of radiation in producing moisture in the steam is not serious; but when the engines are working with a small load, the amount of moisture produced by radiation is much more considerable in proportion to the weight of steam used. This is no doubt one cause of inefficiency when engines are worked with small loads.

Conclusion.—Generally the author thinks that the wiredrawing calorimeter without separator is the most convenient and accurate for steam with less than about 2 per cent. of moisture. For steam containing more moisture, the separating calorimeter without wiredrawing apparatus is accurate enough and convenient. The use of the separator and wiredrawing calorimeter combined is more troublesome, especially if, as is desirable, a condenser is also used to determine the amount of steam passing through the separator. In cases where there is much priming, it would seem best to take the whole of the steam through an ordinary steam separator, measuring the amount of water trapped; and then to test by a wiredrawing or separating calorimeter the dryness of the steam after passing the separator. In priming, much of the water probably flows along the bottom of the pipe, and it appears impossible that a sample can be obtained containing an average proportion of steam and water. It is recommended by Professor Carpenter that the sample of steam to be tested should always be taken from a vertical, not from a horizontal steam pipe. No doubt there is rather more tendeney for water to flow along the bottom of a horizontal pipe than down the sides of a vertical pipe; but merely taking steam from a vertical pipe does not insure freedom from error, especially if the amount of moisture in the steam is considerable. Variations in tests for wetness are doubtless often due

to the difficulty of getting a true average sample of steam; and it would seem that errors are generally in the direction of underestimating the amount of moisture.

Discussion.

Mr. Michael Longridge was the first speaker. He said he was glad to see the paper, for he had suggested many years before, at a meeting of the Institution of Civil Engineers, that a prize should be offered upon the subject. The British Association Committee had done but half the work required. Indeed, the most difficult part of the business was to get a fair sample of the steam. Under these conditions he thought that a small apparatus should not be used, but one which would treat all the steam supplied to a steam engine. Referring to the question of velocity of steam and its effect on carrying forward particles of water, he quoted a case of a large boiler having two openings. Steam was being generated at atmospheric pressure; when one opening was uncovered, water was carried with the steam to a considerable height; when both openings were uncovered, steam only issued forth. He had found that with a velocity of 30 feet per second of steam, water was carried over, but at 8 feet per second there was no water carried forward. This was at atmospheric pressure, but he could not say whether it would hold good at higher pressures. Another experiment bearing on the same subject was one made in supplying steam to a cylinder jacket. It was desirable for a certain purpose to supply the steam and drain the water of condensation by the same pipe, which was $1\frac{1}{2}$ inches in diameter. With a velocity of 5 feet per second at 90 pounds pressure, it was found possible to do this, the steam running up and the water coming down; but if the velocity of the steam were increased above 5 feet, the water would not come down past the steam. On the whole, it seemed to the speaker that it was possible to have a calorimeter to act as separator for an engine, and yet not be of extraordinary size.

Mr. Bryan Donkin spoke as to the value of the paper, and suggested that the Institution should appoint a Research Committee to inquire into the question. He made reference to some French experiments of this nature made on locomotive boilers for the Paris, Lyons and Mediterranean Railway Company. A continuous condensing calorimeter, devised by M. Henri, was used for determining the priming water. This method of working is based on the following principle: A certain quantity of steam is passed into a small surface condenser arranged as a calorimeter, and the increase of heat in the condenser is measured in the circulating water. The greater the quantity of water in the steam the less the heat in the circulating water. From the heat coming from the condenser the percentage of water in the steam is calculated. The steam in the condenser is at atmospheric pressure. The steam to be tested is passed inside the tubes and the circulating water outside, the quantity of the latter being arranged to be constant, as is also the quantity of steam, the pressure of steam being kept uniform in the boiler. The steam is completely condensed in the condenser. The temperatures are taken of the steam before entering the condenser, of the water condensed, and of the circulating water in and out. The quantity of circulating water is also measured. The quantity of heat entering is the same as that leaving the condenser. From an equation it is therefore easy to determine the percentage of water brought over by the steam. In the original paper describing the apparatus, which had appeared in the "Annales des Mines," 1894, vol. vi, full details are given, together with a drawing of the apparatus. The method as used by M. Henri was not, the speaker said, probably new, but he considered it better than many others then known to him. When working the locomotive stationary, M. Henri found the steam practically dry, but he is of opinion that, in running locomotives, the steam always contains a certain amount of moisture.

Some German experiments had also been made to which the speaker referred. He placed on the wall a drawing of Gehre's apparatus, in which a sample of steam, after passing through a pipe, is shut in between two valves. Below there is a means of

heating the pipe by a flame. As long as the steam remains saturated, the pressure and temperature will be according to the steam tables. When the steam is heated from below, an evaporation of all water in the steam takes place, the pressure rises, and the steam commences to be superheated. The volume is constant during the operation. Mr. Donkin described the experiment as follows: "The valves are closed and sample of steam obtained, and the pressure of steam read off by the gauge. The lamp is then placed under the pipe, and as soon as the temperature begins to rise, the steam pressure is again noted. From the weights of these equal volumes at the two unequal pressures, the percentage of moisture in the steam can be calculated. If no moisture is present, the pressure will not rise after flames are applied. The greater the percentage of water present, the greater the difference after heating. The valve must be perfectly steam-tight."

Mr. Druitt Halpin said that the Paris, Lyons, and Mediterranean experiments, he had always understood, gave no certain data, and it was concluded that their results were of no value. Mr. Donkin, in replying to this, said that he understood that practical results were obtained. Mr. Halpin continued that he had been told the experiments were on so large a scale as to be unmanageable, and it was said that the whole of the results had been rejected. The author had described a separating method of determining the wetness of steam. Mr. Halpin would suggest another way. In dealing with two fluids of different specific gravity, it might be possible to throw the whole of the water out by a centrifugal method, using mechanical means, such as a fan, for the purpose; in this way the steam and water would be separated.

Professor T. Hudson Beare said he would like to say something in mitigation of the severity with which the author had treated the salt test. It certainly only showed the mean of mechanical priming, and was not influenced by the condensation; but if moisture were present in the steam, the engineer wanted to know whether it was due to mechanical defects in the boiler

or to radiation of heat, and the salt test might still be used for that purpose. In the continuous condensing method described by the author, he would like to know if the gauges were calibrated, as the results depended on their accuracy. Again, in the combined separating and wiredrawing methods, in which the author said no weighing was required, he would like to know whether there was a gauge on the second chamber. Professor Unwin said he would like to answer these two questions at once. It was strictly necessary that there should be a gauge on the second chamber in the last apparatus referred to, unless the opening were very large. The gauges referred to by Mr. Beare were tested, but he would confess that he did not trust to them very largely. Professor Beare, continuing, said he had grave doubts whether a true sample of the steam was obtained by drawing it out in the present method.

He would give a practical example of the fact by the way in which steam was taken from a chamber in the case of the experimental boiler at the University College. This was a locomotive boiler with a wet bottom, and the steam was taken from a perforated pipe bent upwards. They found on a trial that a large quantity of water was being brought over, and on investigation it was discovered that the pipe had been accidentally shifted downwards, so that though the orifice was still above water, it was very near the water level. With the pipe in this position there were 378 pounds of water brought over with the steam per hour; with the pipe in the normal condition the quantity was 154 pounds per hour, the conditions of working in all other respects being the same in both cases.

Mr. Clarkson said that a suggestion might be taken from the practice of taking sample minerals, in getting a true sample of the steam. A rotating hollow cylinder was used, through which the stream of minerals poured, and this was projected on to the point of a cone. In this way a uniform stream of mineral was obtained, and a thorough admixture was therefore secured. He thought some such apparatus could be introduced into a steam pipe, and by inserting a sector the steam could be diverted and led off to be tested.

Captain Sankey, who was the next speaker, said that the specific heat of steam was generally accepted as .48, and Professor Unwin had so taken it; but he would point out that Mr. Macfarlane Gray had questioned the accuracy of that figure, and, in a correspondence with the late Mr. Willans had given a formula for the specific heat of steam. This showed the specific heat to be not a constant, but to vary with the temperature of the steam. The speaker proceeded to illustrate this by use of the theta-phi diagram by aid of the blackboard. It was important to get rid of this question of the specific heat of the steam, and Mr. Macfarlane Gray and the late Mr. Willans had devised a superheater in which the steam was reduced to a lower pressure than in the Barrus calorimeter, and so the difficulty of the specific heat was got over, but it was not possible to use it, as the true temperature of superheated steam could not be ascertained.

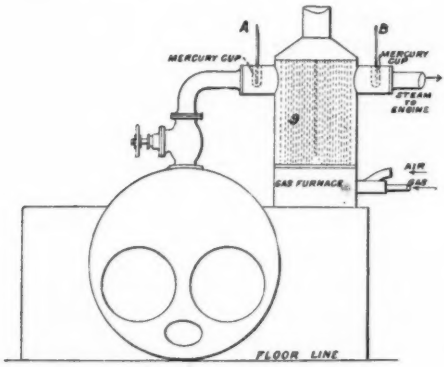
Professor Barr was pleased that the author had treated the salt test so severely, as it was not in any way trustworthy. He had occasion to look into this matter in connection with applying the salt test to a water tube boiler, but he could not tell from what part of the boiler the water came, and the salt would be more in one part than in another. He had suggested the same method of obtaining a true sample of steam as that sketched by the author on the blackboard. His ideas were embodied in the transactions of a scientific society. The difficulty was that the water had greater inertia than the steam, so that the amount of moisture varied with the velocity of the flow. But though he had suggested this method, he was afraid that it would not be of any use, as the water would get to the side of the steam pipe and stay there, and as the nozzle took steam from the center of the pipe, it would not give a fair sample. It would, therefore, be necessary to mix the steam and water, as the previous speaker had suggested; but the best way would be to separate the whole of the water from the steam. He would suggest the use of a spiral pipe taking the water off at the smaller radius.

Mr. Crompton said that when engineers found water in steam pipes, they wanted to know whether it was due to the boilers or

to condensation in the pipes, and he had met with no satisfactory method of determining the problem yet. When they had to deal with a long system of piping they also wanted to know the quantity of water in the steam at various points of the pipe system. If a way could be shown of getting samples of steam, he felt sure that the mean water might be determined by temperature methods, and the electrical engineer would supply means of reading these temperatures accurately. Of late the pyrometer had been so perfected that very trustworthy readings could be obtained by the platinum pyrometer.

Mr. Cawley remarked that Mr. Bryan Donkin had suggested the appointment of a special committee to inquire into and report on the subject of the paper; he would propose another, to deal with the question of the advantages to be derived from moderate superheating of steam supplied to steam engines. He felt so satisfied of the value of these advantages that he predicted that the proposed committee for dealing with the determination of the dryness of steam, and also the one now dealing with the value of steam jackets, would have little work left to do if moderate superheating were generally adopted. However, as such was not yet the case, he would, like other members who had spoken, thank Professor Unwin for his extremely interesting paper. This paper showed that of all the methods mentioned only two could be relied upon. But even these had to deal with only a minute sample of the body of steam taken from the steam pipe, in some way which the experimenter might think best. But it seemed to be generally admitted that this sampling would probably lead to very erroneous results, and he thought it might be easily practicable to apply an apparatus to a medium sized boiler, capable of dealing with the whole of the steam produced, instead of with a minute sample. The arrangement shown by the annexed diagram, which he sketched on the blackboard, would be suggestive of what he meant. The vessel *S* is a superheater with a gas furnace placed in a suitable position near the boiler to be tested. The superheater is fitted with a central baffle plate, reaching from the top

to nearly the bottom, and with a number of vertical tubes. The more or less wet steam from the boiler would pass through the superheater, and the gas supply would be adjusted so that the steam escaping from the superheater was very slightly superheated. This would be shown by the mercury in the thermometer *B* standing slightly higher than in *A*. Then, supposing the evaporative efficiency of the superheater—as a boiler—were known from previous experiment, and also the efficiency of the boiler alone, the relative values of the heat supplied to the boiler furnaces, and the superheater, would supply data for de-



termining the wetness of the steam—after making an allowance for the slight amount of superheat in the steam leaving the superheater. There might be objections, he said, on the score of economy, to the use of coal gas for the superheater, but he pointed out that for a brief experiment this would not be serious. For, taking coal gas at 2s. 8d. per 1,000 cubic feet, the cost per pound would be about 1d., and, having regard to the higher calorific value of coal gas, the cost of gas per thermal unit would only be about seven times that of coal, which was not a serious matter in a short experiment.

Mr. William S. Lockhart referred to the device for sampling steam illustrated by Professor Unwin. The speaker point-

ed out that if the orifice of the connecting nozzle were secured and placed in the middle of the steam pipe, as shown, the sample would not be a fair one, as it would take steam only from the middle of the pipe, whereas the moisture traveled along the walls of the pipe. He suggested, therefore, that the end of the nozzle should be flattened so that the orifice would take the form of a long slit extending across the pipe, and in that way the steam would be taken from the entire diameter.

Mr. Jeremiah Head remarked that he was a member of the committee of the British Association already referred to, and he would like to point out that although there were several members on the committee, the actual work was done by the author of the paper now being discussed, and that the whole credit of the investigation was therefore due to Professor Unwin. The question of separating water and steam, not for the purposes of measurement, but in order to extract the water from the steam, was an old problem. It had been a custom, where long ranges of steam piping were used, to take the main steam pipe into a receiver into which the water would fall. He had known cases where fire had been made at the bottom of this receiver for the re-evaporation of the water. This was a questionable proceeding, and was rarely followed out. What was wanted in an engine was that when the steam passed to the condenser, it should be just saturated steam. That was an ideal case, and in practice it could only be attained by using superheated steam in the high pressure cylinder. He had known cases in which the attempt had been made to get this result by wiredrawing, but he was not aware whether it could be done with advantage. With regard to the question Mr. Lockhart had raised, the speaker questioned whether the water would be present in steam pipes at all parts in equal quantity. It was assumed that in a horizontal pipe water would be at the bottom of the pipe, and for this reason a vertical pipe was used for the purposes of collecting samples, but in this case he doubted whether the proper end was attained. He would, therefore, suggest a telescopic collecting nozzle, which could be pushed in and out so that the collecting orifice could be put in different positions of

the pipe, and in this way the sample collected would be a fairer average.

Mr. J. Mair Rumley said he would like to impress upon the meeting the importance of taking water out of the steam that was to be used in an engine. He would give an instance. In the case of a large compound engine using chalky water in the boiler, they found a very low efficiency in the engine, 26 pounds of steam being required per indicated horse-power, and the indicator cocks being full of water. They put drains on their high pressure valve chest, and extracted 1,200 pounds of water out of 14,000 pounds that were passed from the boiler in the shape of water or steam; this would be an average of about 8 per cent. of water. After this the steam required per indicated horse-power fell to 17 pounds. The fact showed the necessity of a steam separator. Mr. Longridge appeared to think that a very large separator would be required for the purpose of dealing with all the steam passed to an engine, but, as a matter of practice, he (the speaker) was now putting on separators that were no more than five or six times the diameter of the steam pipe. In regard to pipe condensation, he would give an instance showing its importance, and the necessity of keeping the steam pipe as small as possible in order to reduce radiation. A friend of his wished to work a pump at the bottom of a well, and had put on a 2-inch steam pipe, but found that the engine could not be made to revolve at all in consequence of being choked with water. The 2-inch pipe was then taken off and $1\frac{1}{4}$-inch pipe substituted for it, after which, to use his friend's expression, "all went merrily." The steam separator was an important adjunct, and he considered that any one who improved the design would be rendering a service to all steam users.

Mr. Clark said that the subject under discussion was extremely interesting, both from a mechanical and physical point of view. He described Regnault's experiments and the apparatus used in them, and proceeded to remark that since Regnault's day further discoveries had been made. Regnault assumed that all the water in steam would run down into the boiler and not into the calorimeter, and thus may have vitiated the results. The principal

sources of error in calorimeter work were temperature errors, the determination of the latent and specific heat of water and steam, and the capacity of the calorimeter for heat and radiation. The first two might be relegated to the physicist, the other two were common to all apparatus, but they might be eliminated in many cases, and a proper balance maintained between loss and gain. This supposed an ideal calorimeter, but that was a standard that should be aimed at. What was required to be known was the wetness of the steam in the steam pipe, and not that at the end of the tube attached to the pipe, therefore the calorimeter should be in the steam pipe itself. This would give two of the necessary properties of an ideal calorimeter, for the steam as used would be tested, and the calorimeter would always remain at the temperature of the steam. He had designed an instrument of this nature, but one had not yet been made, and he therefore brought forward the suggestion with diffidence. He placed a coil of wire in the steam pipe, and through this passed an electric current; this formed the calorimeter. If such an apparatus were used in air, any fall in temperature would be due to conduction or convection and radiation; whilst if used in dry steam the same conditions would apply. If, on the other hand, the coil were placed in a current of wet steam, the temperature due to the passage of the current could not rise until the water was evaporated. If, therefore, means were supplied for measuring the resistance and the current passing through, there would be a possibility of forming an estimate of the wetness of the steam, for the point at which the temperature began to rise would show that the steam had then reached the saturated condition. To use this apparatus it would be necessary to know the total quantity of steam passing through the pipe. Mr. Clark hoped to carry out the experiments with the apparatus he described, and had already made certain tests, the results obtained being of an encouraging nature.

Professor Unwin, in replying to the discussion, referred to Mr. Longridge's remarks, in which he advocated abandoning small apparatus, and taking the mean of steam passed to the engine for the purpose of examination. It might be, the speaker

said, that this course would have to be followed, but in the meantime he would like to know if Mr. Longridge himself had carried out any experiments in this direction, and if so, whether he found he could approach as closely to accuracy as with the sampling method. He concluded that if Mr. Longridge, for the purposes of a boiler trial, wished to analyze coal, he would take a few grains and draw conclusions as to the whole from analyses made with the small quantity. In like manner, if he wished to gain information as to the composition of chimney gases, he would withdraw small samples and analyze them, and the speaker failed to see why the same course should not be followed with the steam also. Professors Beare and Capper thought he had treated the salt test with scant courtesy, but he had reason to doubt its trustworthiness. He had made salt tests with a boiler without changing the conditions of working in any way, and had found the results vary from 500 to 1,000 per cent. With regard to withdrawing the sample, he feared he had not made himself understood, and had used a word which had led speakers astray. When he spoke of a sample, no doubt he had not chosen the expression well. What he meant was, that in withdrawing a quantity of steam from the pipe for the purpose of testing, there was a liability of the percentage of moisture also taken not being that normal to the steam at the point from which the sample was withdrawn; that is to say, water would accumulate on the withdrawing pipe, and thus give untrue results. He did not fear that the steam in any one part of the steam pipe would be greatly different in humidity to that in any other part; when it was considered that the steam would be rushing through the pipe at 100 feet per second, it would be evident that eddies would be caused and there would be a general mixing up. Under these conditions it was impossible for him to conceive that it was not a homogeneous mixture. It was not, therefore, that the steam in any one part of the pipe differed from that in any other part, so much as the extracting apparatus itself disturbed the current, and thus caused more water to be collected than that properly due to the steam. What he aimed at in designing the extracting nozzle, therefore, was to withdraw the steam without changing

the velocity of flow, and he thought it immaterial in what part of the steam pipe the collecting orifice might be placed. Mr. Cawley had great faith in the future of superheating steam, and Professor Unwin himself thought that it would be largely used in the future. But the application would not come all at once, and, even if it were universal, engineers would want to know what the boiler was doing, and so the test would be with us always. Captain Sankey had spoken of the inaccuracy of thermometers. There were two calorimeters to which he gave preference. In the separating calorimeters, thermometers were not required, so that it was not so much a question of actual temperatures, but agreement between two instruments. At the same time he admitted that there were considerable errors, but these could be got over by calibrating by means of Regnault's tables, if the pressures could be accurately recorded. Mr. McFarlane Gray had carried out some interesting researches to determine the specific heat of superheated steam. Even if these had not an economic value, they were highly interesting and suggestive.

The value he had taken in his paper, viz., .48, was, he maintained, more useful, even if wrong, than Mr. Gray's variable value. It was considered that he had not treated Mr. Willans' experiments, referred to in the paper, with sufficient respect, but that he wished to disclaim. What he had said with regard to the apparatus used by Mr. Willans being the same as the barrel calorimeter was true, although, as explained in the paper, the size was very much greater than that usual. This fact brought him to the subject of what should be the scale of such experiments. Mr. Willans had made them very large in order to eliminate error by using big quantities, but the speaker thought he might have been wrong in his supposition, and in support of this instanced Joule's experiments made with two gallons of water, which were as accurate as those of Mr. Willans with his three tons. Again, the question of convenience came in. With the large apparatus and great volumes of water dealt with by Mr. Willans, one experiment only would be possible under ordinary circumstances; whilst with Hirn's apparatus it would be possible to make half-a-dozen experiments at the same cost, and he thought the mean of these

would be nearer true than the single large trial. Professor Barr had stated that difficulty arose from water traveling in contact with the interior of the steam pipe, and in order to overcome this, he had devised the same apparatus as the speaker had illustrated on the blackboard.

Professor Barr immediately proceeded to knock down the suggestion he had made, by stating that the water traveled at the side of the pipe, and, therefore, the sample taken was not a fair one. His (Professor Unwin's) device, however, was made not with a view of taking steam from any one part of the pipe, but was designed to secure a fair sample by proportioning the nozzle so as to maintain in it a velocity of flow uniform with that maintained in the pipe. Some of the American newspapers had referred to this effect of water in contact with the pipe, but Professor Unwin looked on it simply as a great bugbear. With regard to the position of the calorimeter, it should be put near the boiler if tests for evaporative efficiency were to be made; whilst if tests for engine efficiency were to be carried out it should be placed near the engine. No doubt for engine trials the right thing was to put in a separator, but even then something more was required; and if a calorimeter were placed beyond the separator, then all had been done that was necessary.

[The discussion on this paper is from "Engineering." The nozzle suggested by Professor Unwin for obtaining a sample of the steam is one with its opening turned towards the flowing steam, and so proportioned that the velocity of steam in the nozzle shall be the same as that in the pipe.]

EXPERIMENT TO DETERMINE THE ECONOMIC VAPORIZATION OF GEORGE'S CREEK CUMBERLAND COAL, UNDER CONDITIONS OF ACTUAL PRACTICE ON BOARD THE *DOLPHIN* IN PORT.

By Chief Engineer G. W. Baird, U. S. Navy.

The purpose of the experiment was to determine the economic vaporization of the coal, not alone to estimate its deterioration due to five months' exposure to the weather, but also to determine the efficiency of the boiler.

For this purpose one of the small boilers was used, on its ordinary service in port, *i. e.*, while driving the dynamo, pumping bilges, washing decks, flushing closets, running the distiller and warming and ventilating the ship.

The small condenser with its air and circulating pumps was used; the additional feed water needed was made up from the hydrant on the wharf, but it passed through the feed tank where it was mixed with the warmer water.

The length of stroke of the feed pumps was measured every half hour, and the total number of its double strokes was recorded, from which the piston displacement was calculated for comparison with the feed water delivered, which was actually weighed.

THE BOILER.

The boiler is cylindrical, has two internal cylindrical furnaces, and return tubes above the furnaces; the tubes are of iron, the shell and furnaces of steel. The boiler was built at the Morgan Iron Works in 1883–'84, and has been in use since that time.

Diameter of the boiler, feet	11
Length of boiler, feet and inches	9–7½
Number of furnaces	2
Internal diameter of the furnaces, inches	40
Total number of tubes	216
External diameter of the tubes, inches	2¾
Length of the tubes, feet and inches	6–4
grates, feet and inches	6–9
Width of grates, feet and inches	3–4
Aggregate square feet of grate in both furnaces	45

Aggregate area through the tubes, square feet	7.54
over bridge wall, square feet	9.30
Area through the chimney, square feet	41
Aggregate water heating surface, square feet	1,092
Ratio of heating to grate surface	26 to 1
grate surface to area over bridge wall	4.84 to 1
through tubes for draft	6 to 1
Height of chimney above grates, feet	56.83
Width of top edge of grate bars, inch	1
Width of air space between bars, inch	$\frac{5}{8}$

Feed Pump.—The feed pump is a number 7 Blake pump, having 10 inches diameter of steam end, 6 inches water end, and a nominal stroke of 10 inches.

Feed Tank.—The feed tank is a rectangular iron vessel, well covered with hair felt, and into this tank the air pump, the steam heaters and evaporator drains are led; the feed tank delivers into a filter, but, in these experiments, the filter was not used, as it was not high enough to deliver into the weighing tubs; the feed water was taken directly from the feed tank.

MANNER OF MAKING THE EXPERIMENT.

The coal was all weighed in the fire room as it was brought from the bunker, the height of the water noted in the glass gauge, and the thickness and the quality of the fires noted at the beginning of the experiment; and these were brought to the same condition at the end, as nearly as possible. The feed water was weighed as it flowed from the feed tank into a tub; as each tub became full, and was balanced on the scale, it was emptied into a second tub, from which the feed pump took it. The temperature of each tub of water was noted. The experiment commenced at 1 P. M. on the 26th, and terminated at 1 P. M. on the 28th of March, 1895.

Duration of the experiment in hours	48.
COAL.	
Total pounds consumed	8,000.
of refuse	874.5
combustible	7,125.5
Pounds of coal per hour	166.67
combustible per hour	148.45
coal per square foot of grate surface per hour	3.7

Pounds of combustible per square foot of grate surface per hour	3.09
Percentage of ashes in the coal	10.93
FEED WATER.	
Total pounds pumped into the boiler	71,280.
Mean temperature of the feed water	156.7
steam pressure in the boiler	39.2
ECONOMIC.	
Mean number of pounds of water vaporized per pound of coal	8.91
vaporized per pound of combustible	10,004
vaporized per pound of coal had the feed water been delivered at 212 degrees	9.44
Mean number of pounds of water vaporized per pound of combustible had the feed been delivered at 212 degrees	10.604

REMARKS.

The relatively high economic vaporization is due, largely, to the very low mean rate of combustion. Even when carrying the heaviest load, *i. e.*, the dynamo and ventilating fans, the rate of combustion barely exceeded 5 pounds per square foot of grate surface per hour. The combustion was reduced by closing the ash pit and furnace doors and opening the uptake doors.

The difference between the number of cubic feet of water actually fed to the boiler and the actual piston displacement of the feed pump, even after deducting the slip, was about 9 per cent. The pump had to lift the water about 2 feet, drawing it through a 1-inch pipe for a distance of about 36 inches, though at a mean speed of about four strokes a minute; the high temperature of the feed water caused some vaporization and consequent apparent loss of action. The actual mean stroke of the pump was 8.683 inches. This loss of action, if it may be so called, is not peculiar to any particular make of pump; indeed, there is no essential difference between the water end of steam pumps as now made, but it is due to handling hot water, slip, etc., and must be allowed for in the design. In this test, I find 10 per cent. loss in shortening of stroke due to slowing the pump down, and 9 per cent. loss in addition, as above described. The former of these two losses (the shortening of the stroke) diminishes as the speed of the pump is increased; this variation of speed is much greater in a duplex than a single pump. This uncertainty of the feed pump factor makes it impossible for me to rely upon it as any sort of a meter.

MACHINERY OF BRITISH WAR SHIPS.

By Albert J. Durston, Engineer-in-Chief, British Navy.

[From "Minutes of the Proceedings of the Institution of Civil Engineers."]

The accompanying tables and illustrations, with description of the machinery, are from a paper entitled "The Machinery of War Ships," read before the Institution of Civil Engineers, by Mr. Albert J. Durston, Engineer-in-Chief of the British Navy, the arrangement of the subject-matter being slightly changed; and refer to sixty-four vessels built previously to the Naval Defence Act of 1889, and to seventy built under that act. The latter vessels comprise ten first class battleships, nine first class cruisers, twenty-nine second class cruisers, four third class cruisers, and eighteen torpedo gunboats. Of the hulls, thirty-two were built by contract, and thirty-eight in the Royal dockyards. Of the machinery, that for sixty-three vessels was supplied by contract, and that for seven, consisting of three second class cruisers, one third class cruiser, and three torpedo gunboats, was built in the Royal dockyards.

In Table A are given the particulars of the machinery of nine battleships built before the Naval Defence Act, and of ten others built under that Act. The first six first class, and the *Hero*, a third class battleship, were tried in 1884–'87; the remaining ones in 1893–'94.

First in order (Table A) come seven battleships, six of the first class and the *Hero* of the third class. The trials of these vessels were made in 1884–'87. The engines are vertical in each case, and the boilers are placed back to back against a middle-line bulkhead, and are fired athwartships.

The six first class battleships are fitted with vertical three-cylinder compound twin-screw engines, with one high-pressure and

two low-pressure cylinders in each set, all of which* run at about 100 revolutions per minute with a stroke of 3 feet 9 inches, giving a piston speed of about 750 feet per minute. All, except the *Anson*, are fitted with twelve single-ended three-furnace boilers, with separate combustion chambers, loaded to 90 pounds per square inch. The *Anson* has eight single-ended four-furnace boilers, each fitted with two combustion chambers and loaded to 100 pounds per square inch.

The *Hero* is fitted with vertical two-cylinder compound twin-screw engines, running at 109 revolutions with a stroke of 3 feet, giving a piston speed of 654 feet per minute. This vessel has eight single-ended three-furnace boilers, loaded to 90 pounds per square inch, with separate combustion chambers. The ratio of high- to low-pressure cylinder volumes in all these seven battleships is 1 : 4. The boiler tubes are of iron, with the exception of those of the *Collingwood*, which are of brass. It may be said that since 1881 the use of brass tubes in the Royal Navy for the boilers of large vessels has been practically abandoned.

In 1889–'90 the first-class battleships *Nile* and *Trafalgar* were tried. They have vertical twin-screw triple-expansion engines, running at about 95 revolutions per minute, with a stroke of 4 feet 3 inches, giving a piston speed of about 800 feet per minute. They have six single-ended four-furnace boilers, loaded to 135 pounds per square inch. Each boiler of the *Nile* has one combustion chamber, common to the four furnaces, and steel tubes, while each boiler of the *Trafalgar* has two combustion chambers, two furnaces in each, and iron tubes.

The battleships of the Naval Defence Act (Figs. 1 and 2 and Table A, vessels bracketed *A*) have engines of the vertical three-cylinder triple-expansion type, the ratio of the volumes of the high to the low-pressure cylinders being 1 : 4.84. The revolutions at the full power for which the engines are designed, viz: 13,000 I.H.P., are 108 per minute, the stroke being 4 feet 3 inches, giving a maximum piston speed of 918 feet per minute. Having in view, however, the ordinary rates of steaming, when only a small por-

* Except those of the *Collingwood*, which run at 95 revolutions per minute with a stroke of 3 feet 6 inches, giving a piston speed of 665 feet per minute.

tion of the full power is developed, it was deemed advisable to arrange the valve gear for a maximum of 11,000 I.H.P. while keeping up the maximum speeds for which the engines were designed; and most of the vessels, as will be seen from the table, carried out their full power trials arranged in this manner. The *Royal Sovereign*, however, was tried as originally designed, and developed the 13,000 I.H.P. specified. The natural draft power is 9,000 I.H.P. The boilers in these vessels are eight in number, of the single-ended return-tube type, having four furnaces in each, uniting in two, three, or four separate combustion chambers.* These boilers, two in each water-tight compartment, are placed with their backs towards the longitudinal division, and are stoked from the wings.

In the two battleships bracketed *B* the cylinder ratio is 1 : 5.37, while the piston speed at full power—13,000 I.H.P.—is reduced to 840 feet per minute. The natural draft power is 9,000 I.H.P. The boilers are generally of the same description, and are similarly arranged. In the *Barfleur* and *Centurion* each pair of furnaces is led into a common combustion chamber.

Table B gives the particulars of the machinery of seven first class belted cruisers of the *Australia* class, tried in 1887–'88, of the *Blake* and *Blenheim*, tried in 1891–'92, of the first class cruisers of the *Edgar* type, built under the Naval Defence Act, and of the *Vulcan*, a torpedo depot ship designed about the same time as the *Blake*.

The vessels of the *Australia* class are fitted with horizontal twin-screw triple-expansion engines, running at 110 to 115 revolutions per minute, with strokes between 3 feet 6 inches and 3 feet 8 inches, giving piston speeds of between 770 feet and 840 feet per minute. The ratio of the high- to the low-pressure cylinder is 1 : 4.8. The boilers are cylindrical, double ended, with three furnaces at each end. The *Australia* and *Galatea* have one combustion chamber

* The *Hood*, *Empress of India*, *Repulse*, *Royal Sovereign* and *Ramillies* have one combustion chamber to each pair of furnaces. The *Royal Oak* has a separate combustion chamber to each wing furnace, the two middle furnaces uniting in a common combustion chamber. In the *Resolution* and the *Revenge* each furnace has a separate combustion chamber.

common to the six furnaces; the remainder have three combustion chambers, a furnace from each end leading into each chamber. The steam pressures employed vary between 130 pounds and 137 pounds per square inch. These were the first triple-expansion engines fitted in the navy, and were adopted chiefly at the suggestion of the late Dr. Kirk; although the *Victoria* and *Sans Pareil* had been specified to have triple-expansion engines, but the orders had not been placed. It will be seen from the table that at this period steel boiler tubes were becoming more general, and their adoption instead of iron tubes may be dated 1885–'86.

In the heavily-powered cruizers *Blake* and *Blenheim*, tested in 1891–'92, four sets of engines of the vertical twin-screw triple-expansion type are fitted; two sets driving each shaft, every set being placed in a separate water-tight compartment. The revolutions at full power are 105 per minute, the stroke being 4 feet and the piston speed 840 feet per minute. The main boilers, six in number, are double-ended with four furnaces at each end. Those of the *Blake* have one combustion chamber common to the eight furnaces, divided by brickwork into six compartments, the two center furnaces from the same end leading into one compartment, and the wing furnaces leading separately into the others. The boilers of the *Blenheim* have four combustion chambers, two contiguous furnaces uniting in each.

In the first class cruisers built under the Naval Defence Act, the engines, of the three-cylinder vertical triple-expansion type, are arranged abreast in separate water-tight compartments, divided longitudinally by a bulkhead (Figs. 5 and 6). The boilers are arranged with stokeholds athwartships, separated from each other and from the engine rooms by transverse coal blocks. The dimensions of the cylinders are identical with those of the battleships (except the *Barfleur* and *Centurion*), so that the ratio of cylinder volumes, high to low pressure, is 1 : 4.84; but the revolutions at the full power, 12,000 I.H.P., are only 100 per minute. This, with a stroke of 4 feet 3 inches, gives a piston speed of 850 feet per minute. As in the battleships, the propellers and valve gear, in all except the *Edgar*, *Hawke* and *Grafton*, were subse-

quently arranged for a maximum power of 10,000 I.H.P.—the natural draft power in this instance—so as to enable them to steam more economically at the low powers usual when cruising. The *Edgar*, *Hawke* and *Grafton* made their full power trials at the powers for which they were originally designed, and satisfactorily developed these powers. In the vessels tried at the lower powers, as well as in the battleships similarly dealt with in this respect, the higher power could at any time be developed by suitable modifications in the valve gearing and propellers. The vessels bracketed *C* have four double-ended eight-furnace boilers, and one single-ended boiler for auxiliary purposes, placed in a recess at the forward end of the forward stokehold. Those bracketed *D* have eight single-ended four-furnace boilers. The furnaces are similarly arranged with respect to the combustion chambers in all the main boilers of these vessels, *i. e.*, two contiguous furnaces unite in a common combustion chamber.

The *Vulcan*, torpedo depot-ship, is also fitted with two sets of vertical triple-expansion engines, running at 100 revolutions per minute, with a stroke of 4 feet 3 inches, giving a piston speed of 850 feet per minute. The boilers, loaded to 155 lbs. per square inch, are similar to those of the *Blake*, except that they have three furnaces at each end. A single-ended auxiliary boiler is also fitted as in the *Blake* and *Blenheim*, but of larger size, to meet the requirements of the hydraulic boat-lifting appliances.

Table C gives the data for second class cruisers, the four of the *Mersey* type having been tried in 1885–86, and the others built under the Naval Defence Act.

The *Mersey* and *Severn*, *Thames* and *Forth*, are fitted with horizontal two-cylinder compound twin-screw engines, running at about 120 revolutions per minute with a stroke of 3 feet 3 inches, giving a piston-speed of 780 feet per minute. They have six three-furnace boilers, of the cylindrical direct-tube type, fitted in a fore-and-aft line, and loaded to 110 lbs. per square inch. With the single exception of those of the *Thames*, which have steel tubes, all these boilers have iron tubes. The ratio of cylinders is 1 : 2.8 and 2.9, the size of the low pressure cylinders being small for the full power. Thus, although not economical at full power,

yet at the reduced powers developed on ordinary service a fair ratio of expansion can be used, and in addition there is a certain amount of weight saved—an important item in vessels of this class.

In the succeeding twenty-nine second class cruisers, which were built under the Naval Defence Act, the machinery and armor-protection is arranged similarly to that in the first-class cruisers (Figs. 7 and 8). The engines are of the vertical three-cylinder triple-expansion type, the ratio of cylinder volumes being 1 : 4.88 in those bracketed *E*, and 1 : 5.03 in the remainder. The revolutions at the full power for which these engines are designed—viz., 9,000 I.H.P.—are 140 per minute, the stroke being 3 feet 3 inches, giving a piston speed of 910 feet per minute. The natural draft power of these vessels is 7,000 I.HP. The vessels bracketed *E*, as also the *Æolus* and the *Brilliant*, have three double-ended boilers and two single-ended boilers, with six and three furnaces in each respectively. The object of substituting two single-ended boilers for one double-ended is to provide boilers of suitable dimensions for auxiliary purposes. In the remaining vessels of this class eight single-ended three-furnace boilers are fitted. This subdivision of the boilers has certain advantages, which are considered to more than compensate for the additional weight involved, viz., absence of leaky shell joints, which occur in the long boilers by the racking strains set up; the ability to divide the boilers more nearly in the proportion of the power required to be developed; and, further, the spreading of the wear due to steaming for auxiliary purposes over the whole of the boilers rather than confining it to one or two of them.

In table D, which comprises third class cruisers, the last four are the only ones built under the Naval Defence Act. The *Scout* and *Fearless* have horizontal twin-screw two-cylinder compound engines running at 150 revolutions per minute at full power. The boilers are four in number, of the cylindrical direct-tube type, arranged in two compartments. The working pressure is 120 pounds per square inch, and the ratio of the cylinders 1 : 3.1.

Next in order are the six twin-screw cruisers of the *Archer* class tried in 1886–'87. These also are fitted with horizontal

compound engines and with four three-furnace boilers of the cylindrical direct-tube type. The pressure was here increased to 130 pounds per square inch, with a consequent change of cylinder ratio of 1 : 3.4. The revolutions are 150 per minute, the stroke is 2 feet 9 inches, and the piston speed 825 feet per minute. The *Racoon*, tried in 1888–'89, has similar boilers, usually described as of the "Navy" type, but is fitted with horizontal triple-expansion engines, the boiler pressure being 140 pounds per square inch. These boilers have steel tubes.

The *Medea* and *Medusa* were ordered in 1887, and tried in 1888–'89. They have vertical twin-screw triple-expansion engines running at 140 revolutions per minute at full power; the stroke is 3 feet 3 inches, and the piston speed is 910 feet per minute. The boilers are four in number, double-ended, with three furnaces at each end, and one combustion chamber common to the six furnaces. They were fitted with iron tubes, and work at a pressure of 155 pounds per square inch. The combustion chambers are divided by brickwork into six separate parts.

These were followed in the same year by the *Marathon*, *Magicienne* and *Melpomene*, tested in 1889, which have the last horizontal engines of importance fitted in the Navy. They have twin-screw, triple-expansion engines and four double-ended boilers with three furnaces at each end. There are three combustion chambers to each boiler, one furnace from each end leading into the same chamber. The revolutions are about 140 per minute, the stroke is 3 feet, and the piston speed is 840 feet per minute. The steam-pressure is 155 lbs. per square inch.

About a year later the five cruisers of the *Katoomba* class were ordered for the protection of the floating trade in Australian waters, and were tried in 1890. In these vessels vertical engines are fitted as in all subsequent ships. The piston speed is lower than that in the preceding vessels, ranging from 800 to 815 feet per minute, the stroke being 2 feet 9 inches. There are four double-ended boilers with two furnaces at each end, and with one combustion chamber common to all four furnaces. In 1890 the four smaller cruisers, *Barrosa*, &c., were tested, having engines and boilers of the same type, but with only two instead of four boilers.

In 1888, the third class special-service cruisers, *Barham* and *Bellona*, were ordered. In these vessels, to obtain the high power required with a limited weight of machinery, a modified type of torpedo-boat engine with locomotive-boilers was employed. The engines are of the triple-expansion type, have a stroke of 2 feet 3 inches and were designed to run at 220 revolutions per minute at full power, the piston speed being 990 feet per minute. The boilers are six in number, and are wet-bottomed with a single fire-box; the working pressure is 155 pounds.

[This machinery was designed to develop 6,000 I.H.P. at the above mentioned piston speed.—Ed.]

In the four third class cruisers of the Naval Defence Act (end of Table D) a similar arrangement of machinery is preserved to that in the first and second class cruisers, except that, for the transverse coal-blocks separating the compartments, watertight divisional bulkheads are substituted. The engines are of the vertical twin-screw three-cylinder triple-expansion type, having a ratio of cylinders, high-to-low-pressure, of 1 : 4.97. At the full power for which the engines are designed—7,500 I.H.P.—the revolutions are 160 per minute, the stroke being 2 feet 9 inches, giving a piston speed of 880 feet per minute. The natural draft power is 4,500 I.H.P. They are fitted with four double-ended boilers, having two furnaces at each end, and a separate combustion chamber to each furnace. The boiler pressure is 155 pounds.

Table E relates to the type known as torpedo gunboats, the first of which was the *Rattlesnake*, tried in 1887. These vessels are fitted with vertical triple-expansion engines of torpedo-boat type, and with four wet-bottomed locomotive boilers. The fire box of the *Rattlesnake* is almost completely divided by flat water legs into two fire boxes. The other three vessels of this class are the *Grasshopper*, the *Sandfly* and the *Spider*, which were tested in 1889. The boilers of the latter two are like those of the *Rattlesnake*, but have an undivided fire box, whilst those of the *Grasshopper* are a modification of the type known as "pinnace" boilers; that is, the fire boxes are circular instead of the usual locomotive form. The engines have the same stroke as those of the *Rattlesnake*, viz: 1 foot 6 inches, but they make 300

revolutions, those of the latter being 310 per minute. The piston speed is thus 900 feet against 930 feet per minute in the *Rattlesnake*. The steam pressures employed are: *Rattlesnake*, 140 pounds; *Sandfly* and *Spider*, 145 pounds, and *Grasshopper* 150 pounds per square inch.

The *Sharpshooter* class, tested in 1889–'90, are larger and more powerful, the stroke being increased to 1 foot 9 inches, but the revolutions are only 250 per minute, giving a piston speed of 875 feet per minute. The wet-bottomed locomotive boiler is used in all of the eleven vessels of this class. The number of boilers is four and each fire box is undivided.

[The original design for this machinery contemplated 4,500 I.H.P.—ED.]

In the torpedo gunboats of the Naval Defence Act, Fig. 3, the engines are arranged as in the cruisers, viz., abreast in watertight compartments divided by a middle-line bulkhead, while the boilers are arranged with stokeholds athwartships in two watertight compartments, which in those bracketed *F* are situated forward of the engines. In those bracketed *G*, the two stokehold compartments are placed one forward and one aft of the engines, an arrangement which facilitates supervision and reduces vibration. The engines are of the three-cylinder triple-expansion type, with a cylinder ratio of 1 : 5.37. The revolutions at full power, 3,500 I.H.P. (*Speedy*, 4,500 I.H.P.), are 250 per minute, and the stroke is 1 foot 9 inches, the piston speed being 875 feet per minute. The natural draft power is 2,500 I.H.P. The boilers in all these vessels, except the *Speedy*, are of the locomotive wet-bottomed type, four in number, having (except in the *Gossamer*) the furnace divided in the middle by a water division. In the *Gossamer* no such water division exists, the boilers being similar to those of this class built prior to the Naval Defence Act. The tubes in the boilers of the *Gossamer* and *Gleaner* are smaller than those in the later vessels. In the *Speedy*, Fig. 4, eight Thornycroft water-tube boilers are fitted, four in each compartment, arranged two abreast, with the stokehold dividing them.

The steam pressure is 155 pounds in all except the *Speedy*, when the working pressure is 210 pounds.

The general characteristics of the machinery of the vessels built under the Naval Defence Act having been briefly described, a few comparative remarks may be of interest. As regards steam pressure, it was not considered advantageous to exceed 155 pounds per square inch, as employed in the *Blake*, the *Blenheim*, and in several other vessels ordered prior to the date of the Act; and this pressure is now general in the naval service for boilers of the water-tank type. In the water-tube boilers of the *Speedy* a boiler pressure of 210 pounds per square inch is employed.

An analysis of the weights of machinery and boilers shows that in the eight battleships bracketed *A* the machinery as a whole is lighter per horse power than in the six battleships of the *Admiral* class built prior to the Naval Defence Act, and this increase in horse power per ton is slightly greater when comparing the boiler weights alone. The auxiliary machinery is considerably more powerful in the battleships last constructed than in the vessels of the *Admiral* class, and is therefore heavier; consequently the saving in weight, arising from the employment of triple-expansion engines and a higher steam pressure, is even greater than the figures appear to show. In addition to this saving in weight, there is considerable economy in coal consumption in the later vessels as compared with the earlier type, which had compound engines.

Comparing the *Barfleur* and *Centurion* with the earlier vessels, it will be observed that the saving in weight is less pronounced. This arises particularly from the nature of the service required of these vessels, which are intended for employment on distant stations, where the duration of steaming will be greater and the facilities for repairs less.

In comparison with the weights of that of the *Nile* and of the *Trafalgar*, the machinery of the battleships built under the Naval Defence Act is seen to be heavier. Some of this difference is explained by the heavier auxiliary machinery fitted in the latter vessels, but it must be mainly attributed to the increased weights of the boilers provided, to secure greater subdivision of the boiler power, increased facilities for access and repair, and greater durability.

Comparing the first class cruisers built under the Naval Defence Act with the seven previously referred to, tried in 1887–'88, it will be observed that a slight increase in power for tonnage is shown at the natural draft powers, which is not maintained when the forced draft powers are compared. This arises chiefly from the altered ratio which exists between the natural draft and forced draft powers in the two instances. The object aimed at in the design of the first class cruisers built under the Naval Defence Act, was to maintain a high continuous steaming power, less regard being paid to the possible performances for short periods under forced draft.

A comparison of the weights of the second class cruisers (Table C) built under the Naval Defence Act with the five third class cruisers bracketed H, Table D, which are the type from which these vessels were developed, exhibits the same general features as the first class cruisers, though somewhat more marked, the horse power at forced draft per ton of machinery, as well as of boilers, being considerably less in the later vessels than in the earlier. This arises from the causes already mentioned in regard to the first class cruisers. It is not proposed to carry this comparison further, as what has been mentioned will serve to show the general principles kept in view in determining the weights of machinery of these later vessels.

The machinery of all the battleships built under the Naval Defence Act is arranged in a similar manner, viz., in six separate water-tight compartments, the twin engines being abreast, and the boilers stoked being athwartships. This is illustrated in Figs. 1 and 2. In all the cruisers, the engines are arranged on a similar plan, but the boilers are in two compartments, stoked in the fore-and-aft direction as illustrated in Figs. 5 to 8. It is, of course, well understood that in the design of the machinery for war ships, special attention has to be paid to requirements arising from the necessity of its protection from damage in action. This generally limits the stroke that can be employed.

As regards the details of the machinery of the Naval Defence Act vessels, space only permits of brief reference to them. The barrels of the cylinders in all except the torpedo gunboats, are

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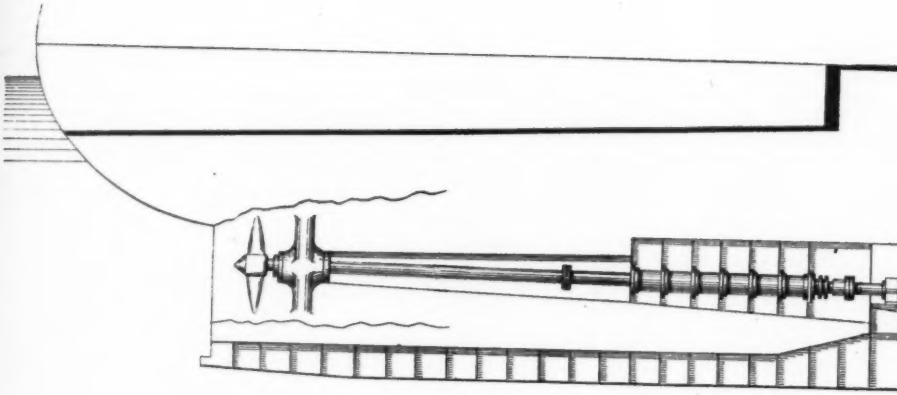
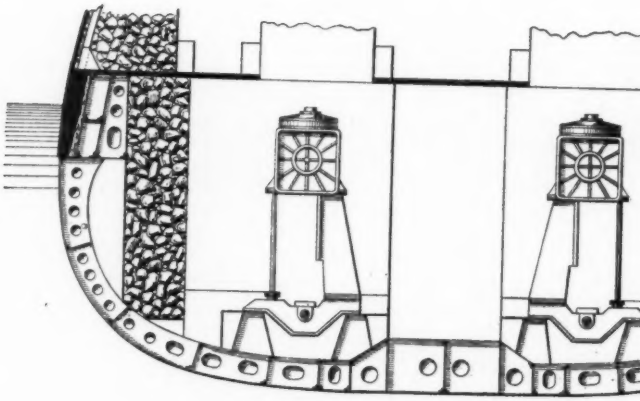
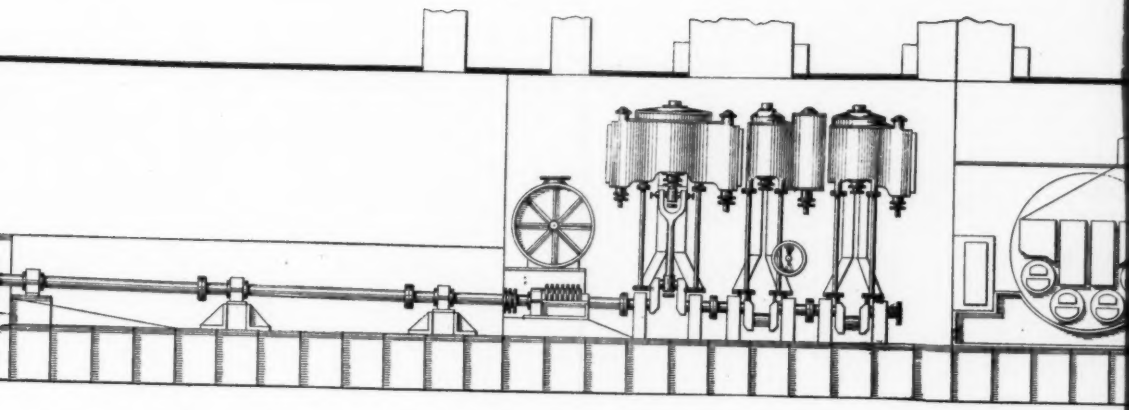


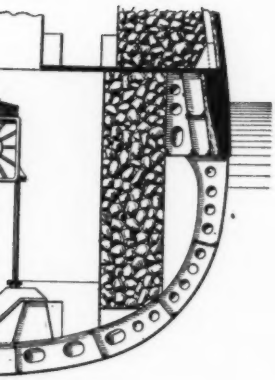


CROSS-SECTION THROUGH ENGINE-ROO

Fig. 1.



Figs 2.



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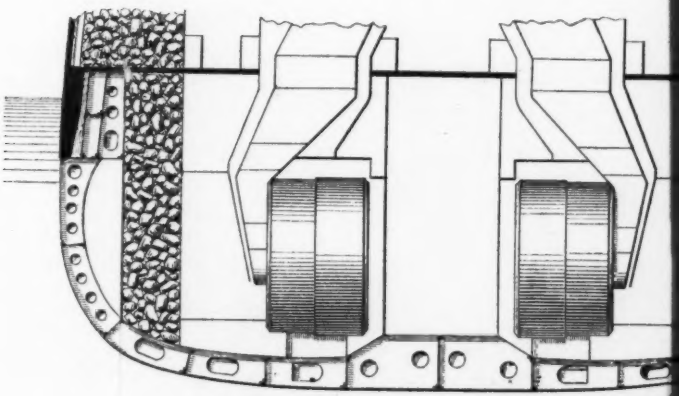

CROSS-SECTION THROUGH BOILER-ROOM.

H. M. S. "RAMILLIES".

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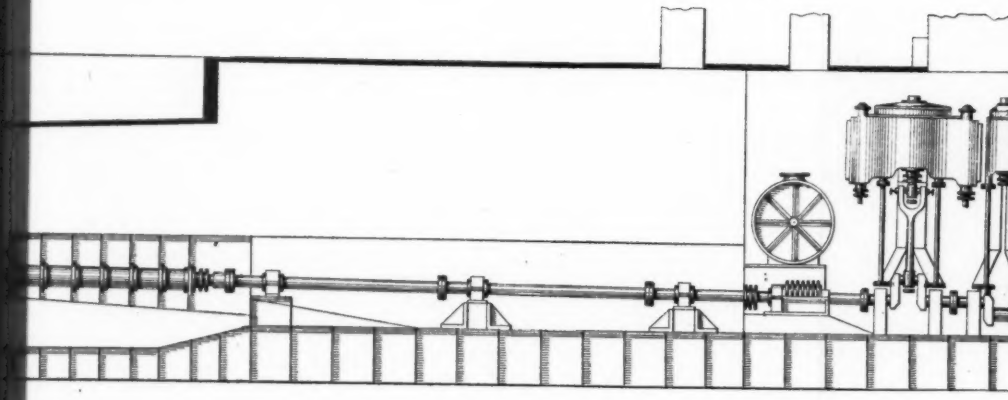
Fig. 1.



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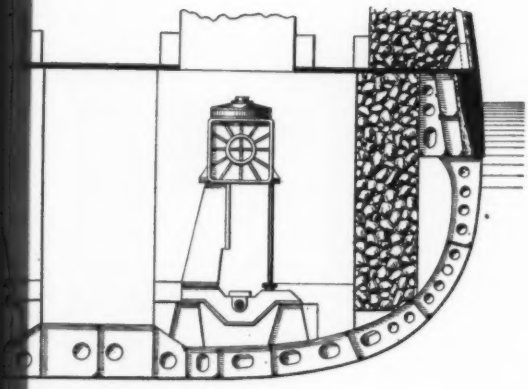

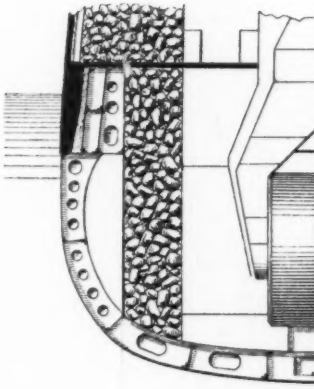

N THROUGH ENGINE-ROOM.

CROSS-S

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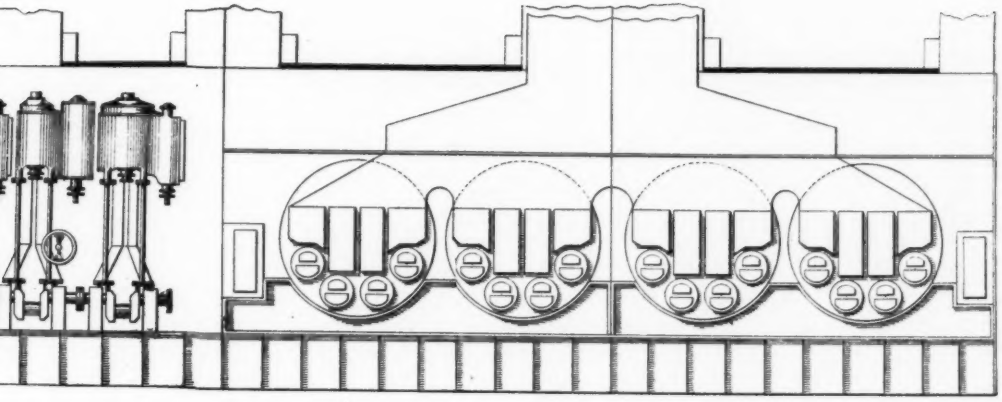
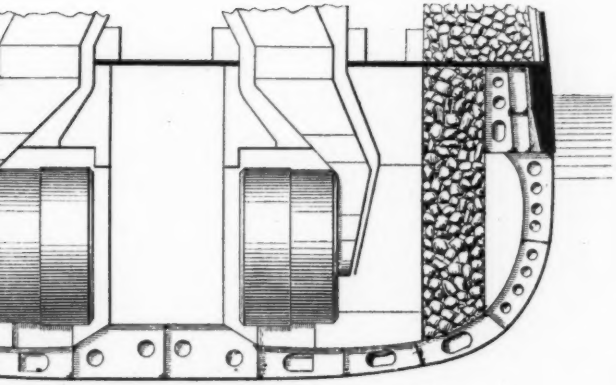


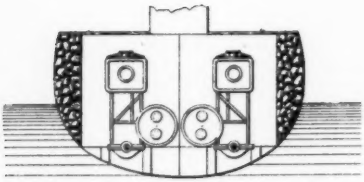


SS-SECTION THROUGH BOILER-ROOM.

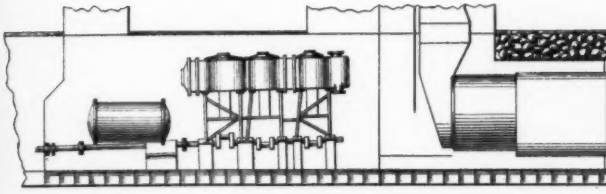
Scale $^1/_{16}$ Inch = 1 Foot.

Feet 10 5 0 10 20 30 40 50 Feet

Figs 3.



CROSS-SECTION THROUGH ENGINE-ROOM.

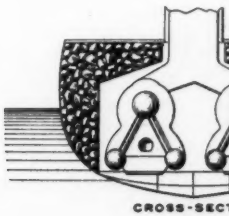


H. M. S. "ONYX" & "R

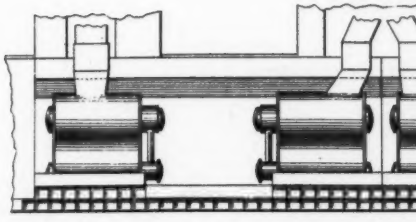
Scale 1/16th Inch = 1 Fe

Feet 10 5 0 10 20

Figs

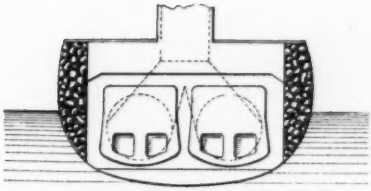


CROSS-SECT

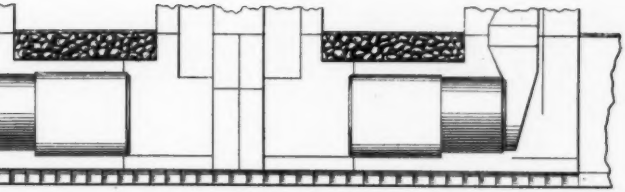


H. M. S. "S

Figs 3.



CROSS-SECTION THROUGH BOILER-ROOM.

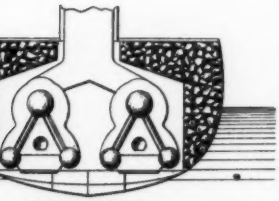


YX" & "RENARD".

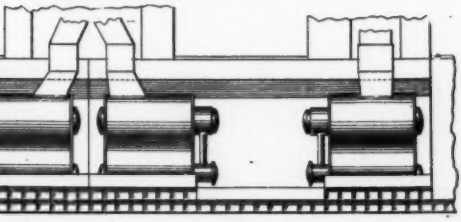
Inch = 1 Foot.

20 30 40 50 *Feet.*

Figs 4.

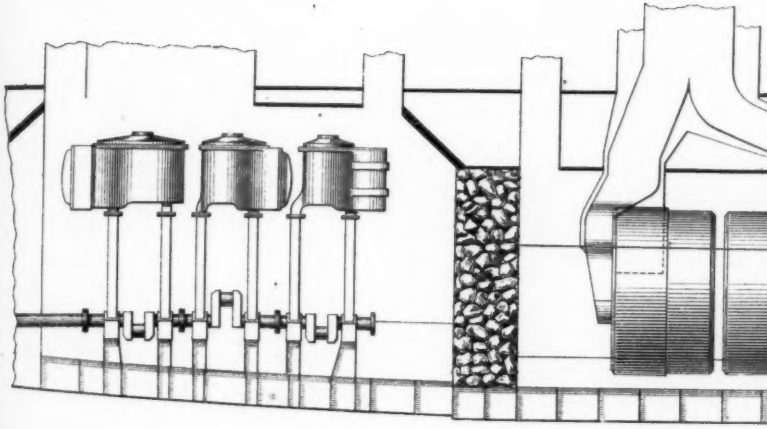


CROSS-SECTION.

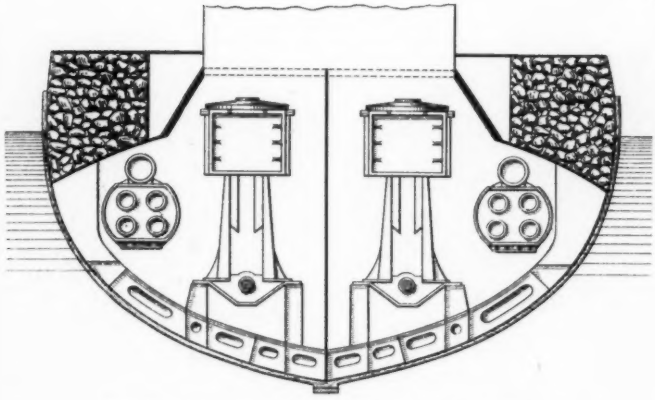


. M. S. "SPEEDY".

Fig: 5.



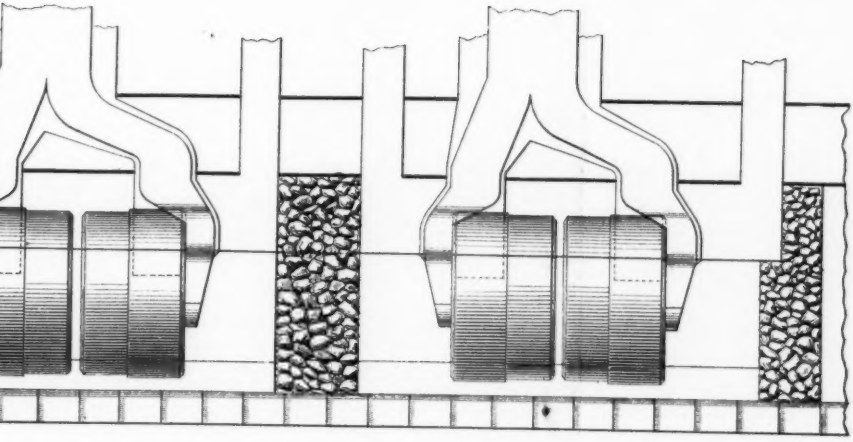
Figs 6



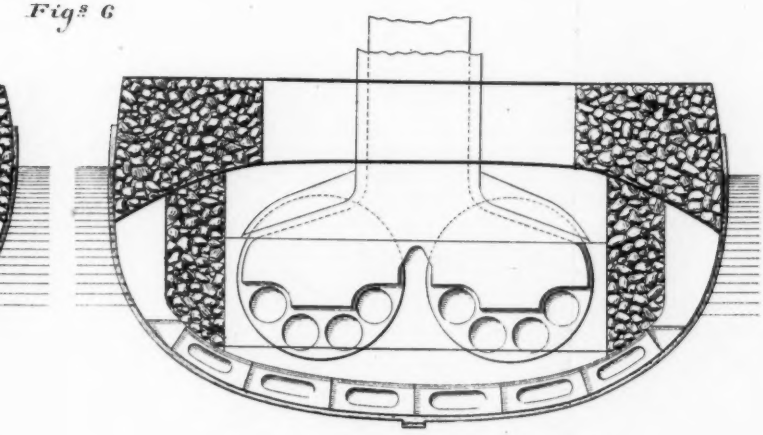
CROSS-SECTION THROUGH ENGINE-ROOM.

H. M. S. "ROYAL ARTHUR".

Fig: 5.

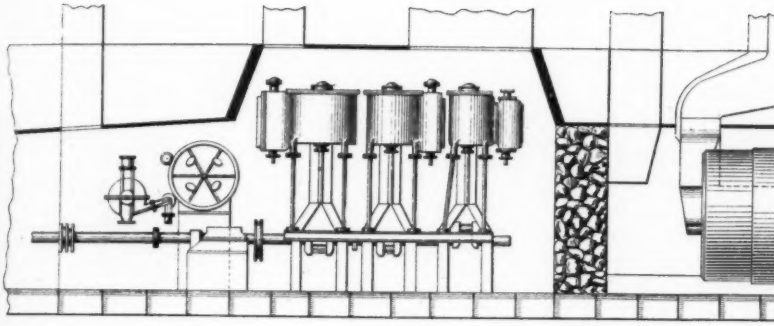


Figs 6

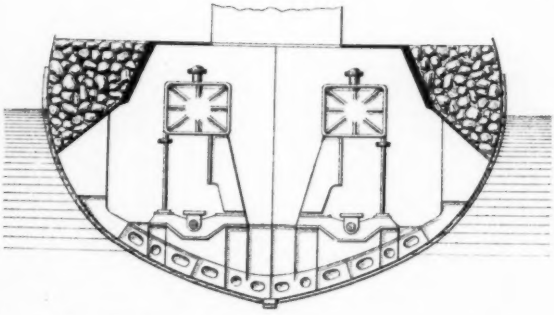


CROSS-SECTION THROUGH BOILER-ROOM.

Feet 10 5 0 Scale 1/16th Inch = 1 Foot. 10 20 30 40 50 Feet.



Figs 8.

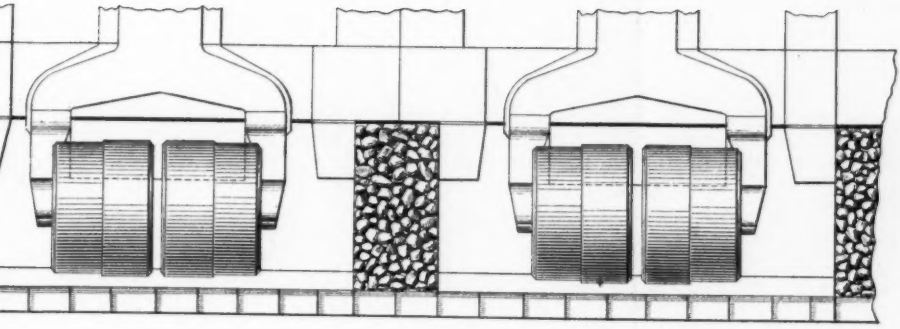


CROSS-SECTION THROUGH ENGINE-ROOM.

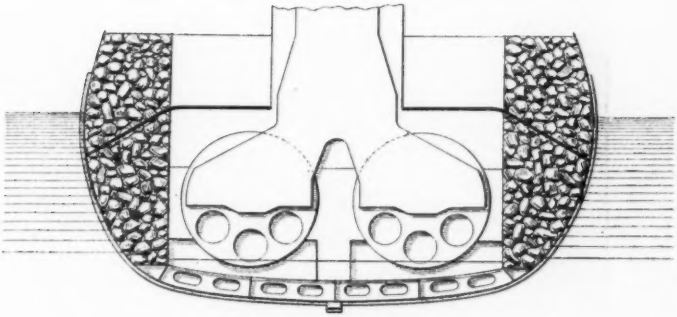
H . M . S . "F

Feet 10 5 0 Scale 10 1/16th Inch = 20

Fig: 7.



8.

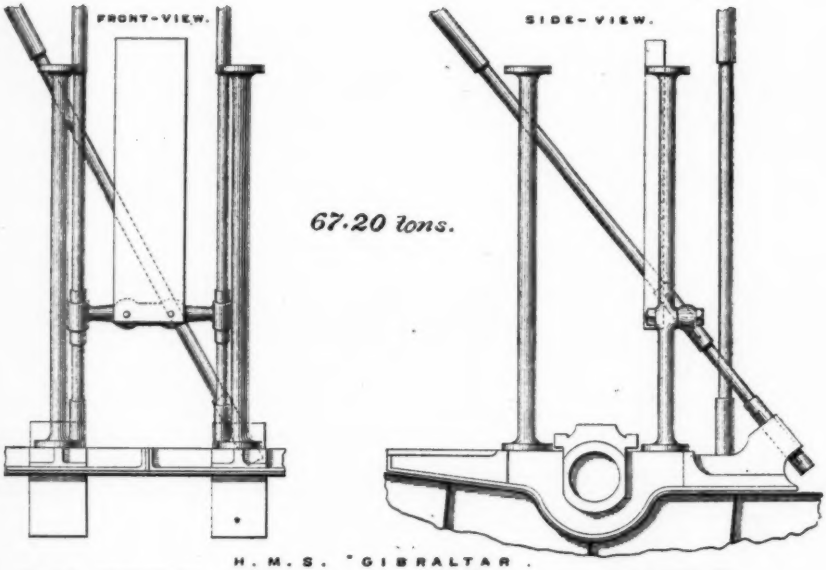


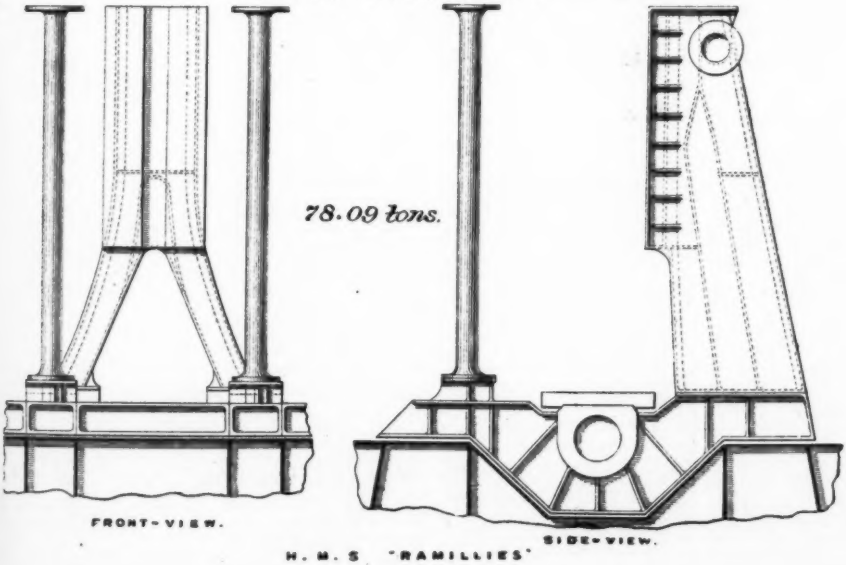
CROSS-SECTION THROUGH BOILER-ROOM.

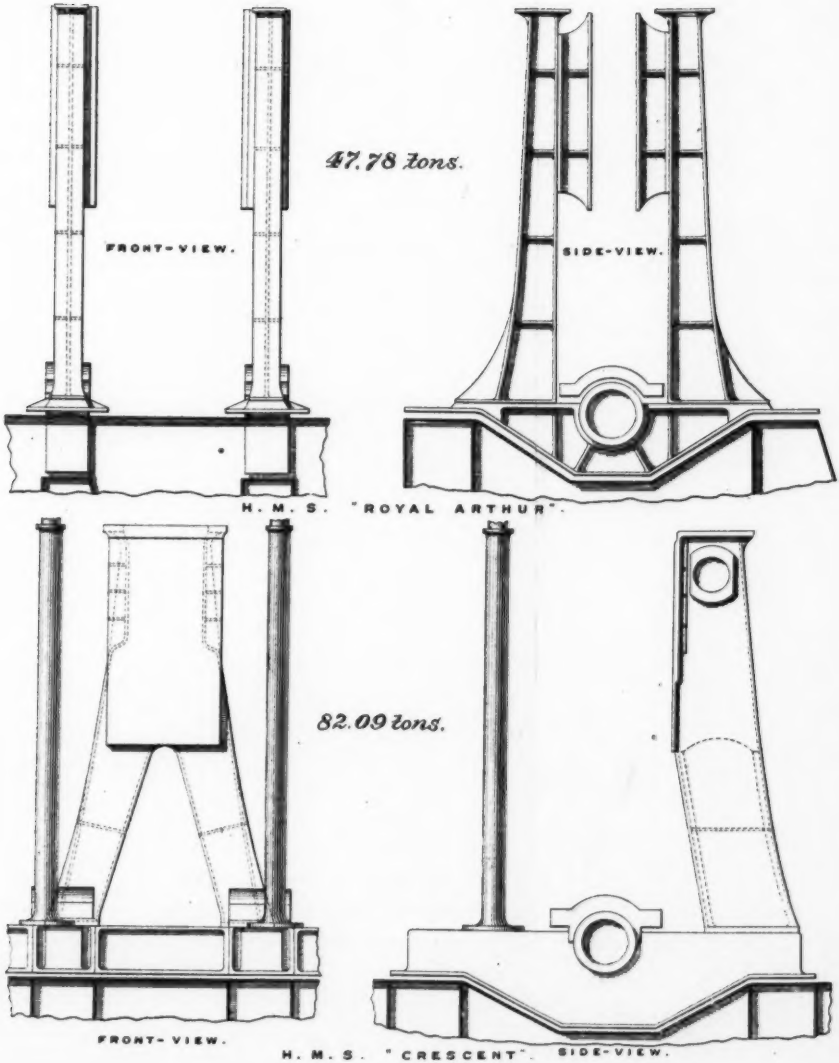
. S. "FORTE"

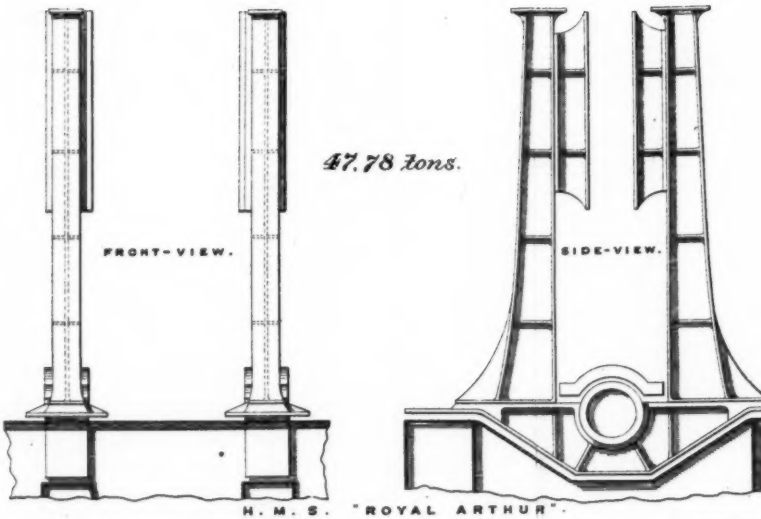
$^{1}/_{16}^{th}$ Inch = 1 Foot.

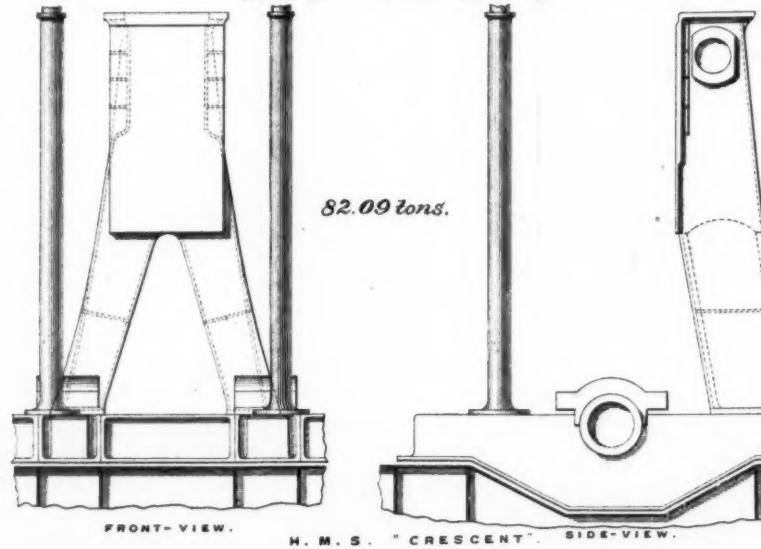
20 30 40 50 Feet.


FRONT-VIEW.
SIDE-VIEW.
67.20 tons.
H. M. S. "GIBRALTAR".

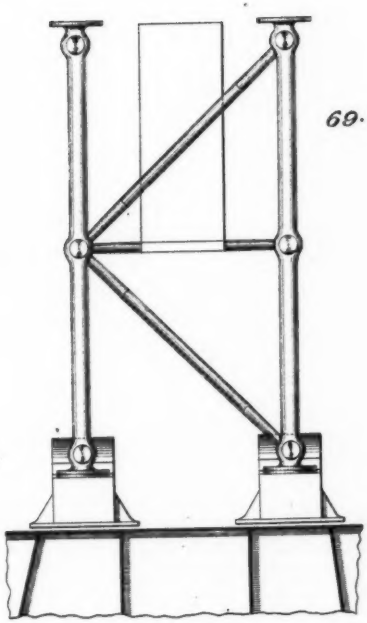

78.09 tons.
FRONT-VIEW.
SIDE-VIEW.
H. M. S. "RAMILLIES"


47.78 tons.
FRONT-VIEW.
SIDE-VIEW.
H. M. S. "ROYAL ARTHUR".
82.09 tons.
FRONT-VIEW.
H. M. S. "CRESCENT". SIDE-VIEW.

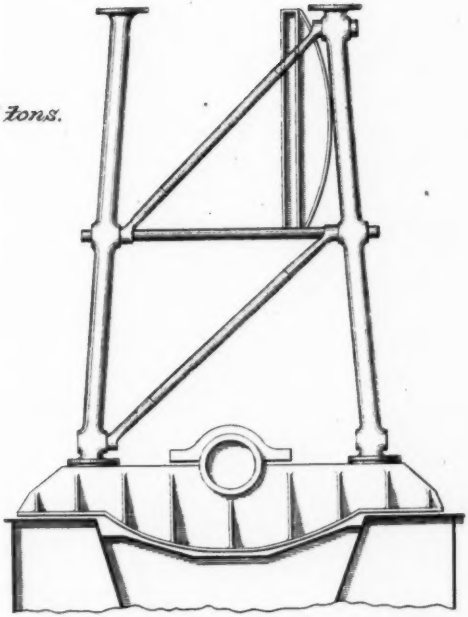

47.78 tons.
FRONT-VIEW.
SIDE-VIEW.
H. M. S. "ROYAL ARTHUR".


82.09 tons.
FRONT-VIEW.
H. M. S. "CRESCENT".
SIDE-VIEW.

69·77 Tons.



FRONT-VIEW.



SIDE-VIEW.

H. M. S. "ROYAL OAK".

Scale 1/4 Inch = 1 Foot.

Feet 5 4 3 2 1 0 5 10 Feet.

jacketed and fitted generally with forged steel liners in the high pressure cylinders, and either forged steel or hard close grained cast iron liners in the intermediate and low pressure cylinders. The engine standards, in all except the torpedo gunboats, consist of one cast steel or iron back column, and two forged steel front columns for each cylinder. There are occasional deviations from this practice, as will be observed by reference to the illustrations of the standards fitted in several of the battleships and first class cruisers, where it is seen that the back and front columns of the *Royal Arthur* are of cast-steel of I section, four to each cylinder, whilst those of the *Gibraltar* and *Royal Oak* are of the forged-steel pillar type with bracing. Table F, column 3, gives the weights of these, including the guide-faces. Although there is nothing in the table to indicate any saving in weight by employing forged steel rather than cast steel for the columns, it is interesting as showing the variation in the weights of these parts, and where a saving in weight might occasionally be effected. The main bearing frames are generally of cast steel, and are in most cases united by distance pieces between consecutive frames. The weights of these are given in column 4, Table F. The pistons are of the steel conical type, having generally in the larger vessels wrought steel junk rings, and for the most part wide single packing rings of either phosphor-bronze or cast iron held out by tempered steel springs. In the torpedo gunboats, Ramsbottom rings of Perkins metal are generally fitted. The crank shafts and propeller shafting are hollow and of forged steel. The condenser cases are of rolled or cast brass. In all, except the torpedo gunboats, separate auxiliary condensers are fitted for taking the auxiliary machinery exhaust. For providing fresh water for "make-up" purposes, reserve fresh water tanks are fitted, in addition to evaporators and distilling plant.

The valve gear in all the seventy vessels consists of the ordinary link motion, working in most cases piston slide valves in the high pressure, either piston or flat slide valves for the intermediate pressure, and usually flat slide valves for the low pressure engines. The air pumps are in all cases worked off the main engines, but separate circulating engines and pumps of the centri-

fugal type are fitted for the condensers. For the purpose of controlling the boilers more readily, especially under forced draft, separate uptakes are fitted to each combustion chamber, with hinged dampers workable from the stokehold floor in each uptake.

For the various auxiliary engines fitted in these vessels, open engines have been generally fitted, instead of the closed type formerly used for the same purposes, also a lower speed of revolution has been adopted. Extra weight has, as already mentioned, been incurred on this account, but it is more than compensated by the greater durability of the machinery. With regard to the boilers of these later vessels, it will be observed that those of the single-ended rather than of the double-ended type have been given increased plate heating surfaces, due to subdividing the combustion chamber, which has the further effect of subdividing the tube plate area. It is considered that these modifications, though increasing the weight of the boilers, ensures stronger and better steaming boilers of a kind more suitable to the varied requirements of Naval service, as previously explained in detail. Representations from the service afloat have been strongly in favor of this change.

As regards the question of leaky boiler tubes, its great importance may make a few remarks excusable. Experience has shown that this defect primarily arises from overheating, due either to forced combustion or to the presence of oily or other deposits on the surfaces exposed to heat; and that this may be mitigated (1) by interposing plate heating surface to absorb the heat before the flames impinge on the tube plate and the tube ends; (2) by the employment of a tube ferrule so constructed as to prevent the flame from impinging on the tube ends and tube plates, and also to prevent conduction of heat to those parts by means of the air space between them and the protecting ferrule; (3) by reducing to a minimum the oil employed for internal lubrication, and by the employment of filters or other processes for its extraction. The employment of all these means has reduced the leaky tube question to one of insignificant character compared with that which it at one time assumed, so that one of the evils consequent on the use of forced draft has been to a large extent successfully

combated. In the smaller vessels a less objectionable difficulty has arisen, viz., the gradual closing up of the ferrules at the tube ends by scoriæ. This appears to depend on the description of coal used, and considerable variation in its amount has been observed. In the return-tube boilers which are fitted in most vessels no trouble arises from this cause, since the tubes are of fairly large diameter. In the locomotive boiler, however, where the tubes are small, some trouble has been experienced; but experiments are being made with the view of minimising it. A different type of tube ferrule, more costly than that mentioned, has also been tried satisfactorily in two boilers of the *Medusa*, and will be fitted in one of the new battleships.

As already mentioned, water-tube boilers for vessels other than torpedo boats have been introduced in the *Speedy*, one of the torpedo gunboats built under the Naval Defence Act. Table E gives the particulars of the performance of this vessel, and the corresponding figures for vessels of the same class will enable a comparison to be instituted. There is seen to be a material increase in the horse power per ton of machinery and boilers. As regards the working capabilities of this type of boiler, immunity from leaky tube ends, the readiness with which steam can be raised, and the absence of all special precautions in their stoking, are points in their favor. On the other hand, the small quantity of water in them, and the rapid evaporation entail considerable attention in feeding; further, their steam space is not great, and care has to be exercised to avoid priming. The small diameter of the tubes precludes the use of salt water for feed, and, further, these boilers exhibit a greater disposition than the water-tank boilers to prime if any cause gives rise to it.

Although not strictly within the range of this paper, possibly a few words on what is now being done in respect of other boilers of the water-tube type may be of interest. In the *Sharpshooter* (Table E), the boilers of the locomotive type have been removed and replaced by eight water-tube boilers of the Belleville type. It cannot be said that any increase in horse power per ton has accrued on this account, but an advantage as regards ability to continuously maintain a high power is shown, arising

more particularly from the fact that there are no tube ends to choke up as in the other type of boiler, and that the whole of the fire side of the heating surfaces is readily cleaned when under way. Particulars of the performances of the *Sharpshooter* fitted with Belleville boilers are given in Table E. It is seen that an increase in weight is involved by their installation above that of the boilers at first fitted, but these latter never generated satisfactorily the required power, even on trials when new. A more just comparison might be instituted between these figures and those of the later vessels of the class in which the boilers have been increased in size and modified in other ways, in order to enable them to attain their intended power satisfactorily. Compared in this way, the boiler weights do not exceed those of the locomotive type. It is to be observed from the air pressures employed, that little or no forcing was resorted to in obtaining the power recorded. In view of the ability of these boilers to maintain a high rate of steaming for considerable periods, and their tactical and other advantages, it has been decided to fit them in the two first class cruisers *Powerful* and *Terrible* now building.

An extended use of the water-tube type of boiler is also being made in the torpedo boat destroyers built and under construction. The table below gives a few particulars of what has been accomplished with these boilers. These figures exhibit the great capability of the boilers to produce steam, but the question of their durability is one that can only be decided after longer experience.

Ship.	Description of boiler.	I.H.P.	Grate area.	Heating surface.	Weight of boiler, mountings, furnace fittings, brickwork, and water to working level.	I.H.P. per ton.
			Sq. feet.	*Sq. feet.*	*Tons.*	
Speedy...	Thornycroft...	4,704	205	17,700	87.45	53.7
Daring..	" ...	4,409	189	8,892	48.5	90.9
Ferret...	Normand.....	4,774	154	8,112	50.7	94 1
Hornet..	Yarrow.....	3,884	172.8	8,216	44.8	86.7

An experiment has been made in another of the vessels built under the Naval Defence Act, viz., in the *Gossamer*, which has

had the Martin system of induced draft fitted to the two forward boilers, the two after boilers retaining their forced draft fittings, thus enabling a comparison to be made between the two systems. The trials have been numerous and prolonged, and experience at sea has also been obtained during the manœuvres of 1893 and 1894. A brief summary of the most important conclusions arrived at may enable an idea to be formed of the merits of the system. The draft can be accelerated equally by either system, but a fan of considerably larger dimensions is required for the induced draft, and this occupies a disadvantageous position by being placed in the uptake. As regards absence of leaky tube ends, no superiority can be claimed for induced draft. The formation of scoriæ at the tube ends occurs apparently to the same extent with both systems. With induced draft, the fitting of a separate fan in the uptake of each boiler places the boilers under better control for cleaning fires, etc. The open stokehold, consequent upon induced draft, enables much freer communication to be kept up with the other machinery spaces. The temperature of the stokeholds is lower, and the stokers work generally in greater comfort; as a consequence, the stoking is better, and this, combined with the maintenance of more perfect control over the fires, helps to reduce the coal consumption, which the trials show to be lower. This system of draft is being fitted to one of the new first-class battleships now under construction, and to another vessel of the gunboat class.

Several of the vessels forming the subject of this paper have been sufficiently long in commission to have made passages at powers approaching their natural draft; and these indicate that the vessels are capable of maintaining, without any special effort and with their own complement of stokers, a high percentage of their natural draft powers. A few of these trials may be cited: The *Royal Sovereign*, on a passage from Plymouth to Gibraltar, maintained 8,180 I.H.P. for seventy-two hours, the coal consumed being 1.84 pounds per hour. The *Royal Arthur*, on a passage from Callao to Coquimbo, maintained 8,821 I.H.P. for seventy-two hours with a coal consumption of 1.85 pounds per I.H.P. per hour. The *Sans Pareil*, on a passage from Malta to Volo, de-

TABLE A.—

	Ship.	I.H.P. on trial, natural draft.	Air-pressure.	I.H.P. on trial, forced draft.	Air pressure.	Weights.		Surfaces.	
						Machinery complete.	Boilers.	Grate.	Total heating.
			Inch.		*Ins.*	*Tons.*	*Tons.*	*Sq. ft.*	*Sq. ft.*
	Benbow	8,662	..	10,853	1.3	1,274	717.2	798	20,500
	Howe	7,730	..	11,725	2.4	1,151	608.4	756	20,726
	Rodney	8,259	..	11,158	1 4	1,159	636 5	756	20.726
	Camperdown	8,605	..	11,739	2.4	1,276	695 5	826	20,500
	Collingwood	8,100	..	9,570	1.1	1,244	736 1	810	20,776
	Anson	8 305	0 2	12,585	1.9	1,149	612.1	797	20,296
	Average	8,277	..	11,271	1.7	1,209	667 6	790	20.587
	Nile	9,106	0.5	12,109	1.7	1,017	537.7	600	19 560
	Trafalgar	8.520	0 5	12,822	2.1	1 013	501.2	659	18,300
	Average	8,813	0.5	12,465	1.9	1,015	519.4	629	18,930
	Hero	4.351	..	6,162	1.0	817	483.9	507	14,377
A	*Hood*	9,539	0 4	11,446	1.0	1,155	583.4	734	20,034
A	*Empress of India*	9,508	0.5	11,625	1.0	1,149	581.1	728	20 034
A	*Repulse*	9.588	0.4	11,314	0 9	1,149	586.6	718	20,034
A	*Royal Sovereign*	9,661	0.4	13 363*	1.6	1,159	597.9	718	20,034
A	*Ramillies*	9,443	0 2	11,571	0.3	1,159	587.7	711	20,132
A	*Resolution*	9,248	0 2	11,402	0.6	1,161	608 1	765	21,178
A	*Revenge*	9.220	0.2	11,536	0.4	1,158	607.2	765	21,178
A	*Royal Oak*	9,235	0.2	11,608	0.9	1,216	610.3	710	20,160
	Average	9,430	0 3	11,500	0.8	1,163	595 3	731	20,348
B	*Barfleur*	9,934	..	13 163	1.7	1,302	668.4	774	22,317
B	*Centurion*	9.711	0 18	13,214	1 5	1,307	660 6	774	22,317
	Average	9,822	0.09	13,188	1.6	1,304	664.5	774	22,317

*Omitted from average.

BATTLESHIPS.

I.H.P. per ton.				I.H P. per square foot of grate.		Heating surface per I.H.P.		Boiler-tubes material.
Machinery complete.		Boilers.		Natural draft.	Forced draft.	Natural draft.	Forced draft.	
N. D.	F. D.	N. D.	F. D.					
						Sq. feet.	*Sq. feet.*	
6.8	8.5	12.0	15.1	10.8	13.6	2.3	1.8	Iron.
6.6	11 8	11.0	19.2	10.2	15 5	2.6	1.7	"
7.1	9.6	12.9	17.5	10 9	14.7	2.5	1.8	"
6.7	9 2	12.3	16.8	10 4	14 2	2.3	1.7	"
6.5	7.6	11.0	13.0	10.0	11.8	2.5	2.1	Brass.
7.2	10.9	13.5	20.5	10 4	15.7	2.4	1.6	Iron.
6 8	9 3	12 4	16.9	10 4	14.2	2.4	1.8	..
8 9	11.9	16.9	22 5	15.1	20.1	2.1	1.6	Steel.
8 4	12.6	17 0	25.5	12.9	19.4	2.1	1.4	Iron.
8 6	12 2	16.9	24.0	14 0	19.7	2.1	1.5	..
5 3	7 5	8.9	12 7	8.5	12.1	3.3	2.3	Iron.
8.2	9.9	16.3	19.6	13.0	15 6	2.1	1.7	Steel.
8 2	10.1	16 3	20.0	13.0	15.9	2.1	1.7	"
8 3	9 8	16.3	19.3	13.3	15 75	2.0	1.7	"
8.3	11.5	16.1	22.3	13.4	18.6	2.0	1 5	"
8.1	9 9	16.0	19.7	13.2	16 2	2.1	1.7	"
8.0	9.8	15.2	18.7	12 0	14.9	2.2	1.8	"
8.0	9 9	15.1	19 0	12.0	15.0	2.2	1.8	"
7.6	9.5	15 1	19.0	13.0	16.3	2.1	1.7	"
8.1	10.0	15.8	19 7	12.9	16.0	2 1	1.7	..
7.6	10.1	14.8	19 7	12 8	17.0	2 2	1.7	Steel.
7 4	10.1	14.7	20.0	12.5	17.0	2.2	1.7	"
7 5	10 1	14.7	19 8	12.6	17.0	2.2	1.7	..

TABLE B.—

	Ship.	I.H.P. on trial, natural draft.	Air pressure.	I.H.P. on trial, forced draft.	Air pressure.	Weights.		Surfaces.	
						Machinery complete.	Boilers	Grate.	Total heating.
			Inch.		*Ins.*	*Tons.*	*Tons.*	*Sq. ft.*	*Sq. ft*
	Australia	5,810	Nil	8,876	1.7	780	413.2	513	15.191
	Orlando	5,628	...	8,739	1.0	795	430.2	548	15.795
	Undaunted	5,682	...	8,670	1.8	792	425 6	548	15.795
	Narcissus	5,353	...	8,589	2.0	770	434.8	535	15 524
	Galatea	5,871	...	9,220	1.1	786	434 3	495	15.191
	Immortalite	6,090	...	8,738	2.0	772	440 0	492	15,190
	Aurora	5,706	...	9,013	1.2	818	457.5	471	15,832
	Average	5,734	...	8,835	1.5	787	433 6	514	15 502
	Blake	14,525	0 4	...	...	1,017	787.2	871	27,192
	Blenheim	14,924	0.2	21,411	2.0	1,013	754.0	1,135	31.073
	Average	14,724	0.3	21,411	2.0	1,015	770.6	1,003	29,132
C	*Edgar*	10,172	0.3	12,550	0.7	1,135	639 8	849	24.879
C	*Hawke*	10,761	0.3	12,521	0.4	1,144	628.5	812	24,879
C	*Endymion*	10,662	0.1	...	...	1,129	654.2	847	25.173
C	*Gibraltar*	10,445	0 1	...	...	1,178	606.9	746	24,754
C	*St. George*	10.585	0 1	...	...	1,155	670 7	847	25,173
D	*Royal Arthur*	10,086	0 4	...	...	1,186	663.1	731	24.828
D	*Crescent*	10,378	...	...	...	1,188	661 1	860	24,784
D	*Grafton*	10,956	0 4	13,483	1.1	1,164	644.7	875	24,880
D	*Theseus*	10,608	0.4	...	...	1,170	650.4	751	24,828
	Average	10 517	0.2	12,851	0.7	1,161	646.6	812	24,908
	*Vulcan**	8,167	0 4	12,062	1.8	988	495 5	564	15,861

* Torpedo depot ship.

FIRST CLASS CRUISERS.

I.H.P. per ton.				I.H.P. per square foot of grate.		Heating surface per I.H.P.		Boiler tubes, material.
Machinery complete.		Boilers.		Natural draft	Forced draft.	Natural draft.	Forced draft.	
N. D.	F. D.	N. D.	F. D.					
						Sq. ft.	*Sq. ft.*	
7.4	11.3	14.0	21.4	11.3	17.3	2.6	1 7	Steel.
7.0	10.9	13.0	20.3	10.2	15.7	2.8	1.8	"
7.1	10.9	13 3	20.4	10.3	15.8	2.7	1 8	"
6.9	11.1	12 3	19 7	10.3	16.0	2.9	1.8	Steel: Stay, Iron.
7.4	11.7	13.5	21.2	11.9	18.6	2.5	1 6	Iron.
7.8	11.3	13.8	19.8	12.3	17.7	2.5	1.7	Steel. Stay, Iron.
6.9	11.0	12.4	19.7	12.1	19.1	2.7	1.7	Steel.
7.3	11.2	13 2	20 7	11.2	17.2	2.7	1.7	
9.3	...	18.4	...	16.6	...	1.8	...	Steel.
9.6	13.8	19.7	28.3	13.1	18.8	2.0	1.4	"
9.4	13.8	19.0	28.3	14 8	18.8	1.9	1 4	
8.9	11.0	15.8	19.6	12 0	14.7	2.4	1.9	Steel.
9.4	10.9	17 1	19.9	13.2	15 5	2.3	1 9	"
9.4	...	16.3	...	12.5	...	2.3	...	"
8.8	...	17.2	...	13.9	...	2.3	...	"
9.1	...	15.7	...	12 5	...	2.3	...	"
8.4	...	15.2	...	13.7	...	2.4	...	"
8.7	...	15.7	...	12.2	...	2.3	...	"
9.4	11.5	17.0	20 9	12.5	15.4	2.2	1.8	"
9.0	...	16.3	...	14.0	...	2.3	...	"
9.0	11.1	16.2	20.1	12.9	15.2	2.3	1.9	
8.2	12.2	16.4	24.3	14.4	21.3	1.9	1.3	Steel.

TABLE C.—

	Ship.	I.H.P. on trial, natural draft.	Air pressure	I.H.P. on trial, forced draft.	Air pressure.	Weights.		Surfaces.	
						Machinery complete	Boilers	Grate.	Total heating.
			Inch.		*Inches.*	*Tons.*	*Tons.*	*Sq. ft.*	*Sq. ft.*
	Mersey	4,515	Nil	6,628	2.0	552	307.9	399	11,711
	Severn	4,603	"	6,335	2.0	552	306.6	399	11,711
	Thames	4,162	"	5,886	1.4	563	311.4	377	11,444
	Forth	3 590	"	5 756	1.9	568	320 2	380	11,393
	Average	4,217	Nil	6,151	1.8	559	311.0	388	11.565
E	*Andromache*	7,234	0.5	9,044	1.3	714	435.5	560	15.389
E	*Apollo*	7.488	0.3	9.226	0.8	715	432.5	560	15,389
E	*Indefatigable*	7,349	0.4	9,047	0.9	735	4 2.2	588	15.725
E	*Intrepid*	7,522	0.4	9.489	1.2	739	420 9	588	15.725
E	*Iphigenia*	7,251	0.2	9.337	0.8	736	418.0	546	15.725
E	*Latona*	7,261	0.4	9.455	1.3	728	433.6	584	15.512
E	*Melampus*	7,684	0.4	9,135	1.2	713	431.7	588	15.512
E	*Naiad*	7.547	0.4	9,261	1.2	703	425.2	584	15,512
E	*Pique*	7.515	0.2	9,258	1.0	751	422.8	574	15.704
E	*Rainbow*	7,879	0.3	9.741	0.7	751	422.8	574	15,704
E	*Retribution*	7,645	0.2	9,367	0 6	751	422.8	574	15.704
E	*Sappho*	7.301	0 3	9,618	0.7	727	420.7	593	15.754
E	*Scylla*	7.614	0.2	9,280	0 8	727	421.0	593	15.754
E	*Sirius*	7,491	0.2	9.281	0 8	742	428.0	602	15 918
E	*Spartan*	7,832	0.3	9.254	0 9	.....	428.0	581	15.918
E	*Sybille*	7.598	0.1	9.524	0.5	726	425.7	604	16,039
E	*Terpsichore*	7.133	0 2	8,825	0.9	718	402.3	573	15,470
E	*Thetis*	7,034	0.5	9 496	0.7	718	402.3	553	15 470
E	*Tribune*	7,523	0 1	9.101	0.5	718	402 3	553	15.470
	Æolus	7,504	0.1	9.315	0.7	.....	417 0	555	15.947
	Brilliant	7,522	0.1	9,180	1.1	732	439.0	556	15 947
	Bonaventure	7.423	0.4	9.365	0 8	797	469.4	553	15,600
	Cambrian	7,164	0.4	9,259	1 0	776	455.0	567	15,600
	Charybdis	7,109	0 5	9.136	1.6	.....	488.3	574	15,304
	Flora	7.187	0 4	9 008	1.3	.....	458.4	580	15,788
	Astræa	7,603	0.4	9.151	1 4	813	466 9	593	15.287
	Fox	7,034	0.3	9,063	1.3	...	460.0	600	15,655
	Forte	*	...	...	...	...	445.2	584	15,600
	Hermione	7.393	0.4	9,264	0.9	795	481.4	573	15 440
	Average	7.437	0.3	9.274	0.9	740	434.4	575	15,641

* Not yet tried.

NOTE.—The weights given in column 7 comprise boilers, funnels, casings, pipes, spare gear, and all fittings in boiler rooms; also water in boilers at working level.

SECOND CLASS CRUISERS.

I H.P. per ton.				I.H.P. per square foot of grate.		Heating surface per I.H.P.		Boiler tubes material.
Machinery complete.		Boilers.		Natural draft.	Forced draft.	Natural draft.	Forced draft.	
N. D.	F. D.	N. D.	F. D.					
						Sq. ft.	*Sq. ft.*	
8.1	12.0	14.6	21.5	11.3	16.6	2.5	1.7	Iron.
8.3	11.4	15.0	20.7	11.5	15.8	2.5	1.8	"
7.3	10 4	13 3	18.9	11.0	15.6	2.7	1.9	Steel.
6.3	10.1	11.2	18.0	9.4	15.1	3.1	1.9	Iron.
7.5	11.0	13.5	19.7	10 8	15.8	2.7	1.8	...
10.1	12 6	16 6	20 7	12.9	16.1	2 1	1.7	Steel.
10.4	12 9	17 3	21.3	13.3	16.4	2.0	1.6	"
9.9	12.3	17.3	21 4	12.5	15.3	2.1	1.7	"
10.1	12.8	17.8	22 5	12.8	16.1	2 0	1 6	"
9.8	12.6	17 3	22 3	13.2	17.1	2.1	1.6	"
9.9	13.0	16.7	21 8	12.4	16.1	2 1	1.6	"
10.7	12.8	17.8	21.1	13.0	15 5	2.0	1 7	"
10.7	13.1	17 7	21.7	12.9	15.8	2.0	1.6	"
10.0	12 3	17.7	21.8	13.0	16 1	2 0	1.7	"
10.4	12.9	18 6	23.0	13 7	16 9	2 0	1.6	"
10.1	12 4	18.0	22 1	13.3	16.3	2.0	1 6	"
10.0	13.2	17.3	22 8	12.3	16.2	2.1	1.6	"
10.4	12 7	18.0	22 0	12.8	15.6	2.0	1.7	"
10.0	12.5	17.5	21 7	12 4	15.4	2.1	1.7	"
...	...	18.3	21.6	13.4	15 9	2.0	1.7	"
10.4	13 1	17.8	22.3	12.5	15.7	2.1	1.6	"
9.9	12.2	17.7	21.9	12.4	15.4	2.1	1.7	"
9.8	13.2	17.4	23 6	12.7	17.1	2.1	1.6	"
10.4	12.6	18.7	22 6	13.6	16.4	2.0	1.7	"
...	..	18.0	22.3	13.5	16.7	2 1	1.7	"
10.2	12 5	17.1	20.9	13 5	16.5	2.1	1.7	"
9.3	11.7	15 8	19.5	13.4	16.9	2.1	1.6	"
9.2	11.9	15.7	20.3	12 6	16.3	2.1	1.6	"
...	...	14.5	18 7	12.4	15 9	2.1	1.6	"
...	...	15.6	19.6	12.4	15.5	2.1	1.7	"
9.3	11.2	16.2	21 7	12.8	15.4	2.0	1.6	"
....	...	15.3	19.7	11.7	15.1	2.2	1.7	"
...	...	...	...	...	...	...	...	"
9.0	11.6	15.3	19.2	12.9	16.1	2.0	1.6	"
10.0	12 5	16.5	20.7	12 9	16.1	2.1	1.6	"

TABLE D.—

Ship.	I.H.P. on trial, natural draft.	Air pressure.	I.H.P. on trial, forced draft.	Air pressure.	Weights.		Surfaces.	
					Machinery complete.	Boilers.	Grate.	Total heating.
		Inch.		*Ins.*	*Tons.*	*Tons.*	*Sq. ft.*	*Sq. ft*
Scout	2,162	Nil	3,370	1 5	291	174.1	207	6,362
Fearless	2,241	"	3,360	2.0	302	185 8	217	6,439
Average	2,201	"	3,365	1 7	296	179 9	213	6.400
Archer	2,220	Nil	3.850	1.0	354	206.8	209	6,836
Brisk	2,614	"	3,816	0.9	354	206.2	209	6,836
Cossack	2,335	"	3,700	1.0	354	208.1	228	6,836
Mohawk	2,577	"	3 398	0 9	352	202.1	228	6,836
Porpoise	2,476	"	3,944	1.0	353	206.2	234	6,836
Tartar	2,554	"	3,824	0.9	355	204.9	228	6,836
Average	2,462	"	3.754	0.9	353	205.7	222	6,836
Racoon	2,647	Nil	4,613	1.5	395	226.6	244	7,878
H *Medea*	6,027	0.5	9,185	2.2	624	373.5	468	12,628
H *Medusa*	6,144	...	9,435	1.9	614	363 8	525	12,628
H *Marathon*	6,530	0.4	8,786	2.3	624	360.2	535	13,616
H *Magicienne*	5.408	0.6	9,280	2 2	627	365.0	535	13 616
H *Melpomene*	6,215	0.4	9,653	1.7	649	369 0	570	13,830
Average	6,065	0.5	9,268	2.0	627	366.3	526	13,264

NOTE.—The weights given in column 7 comprise boilers, funnels, casings, pipes, spare gear, and all fittings in boiler rooms; also water in boilers at working level.

THIRD CLASS CRUISERS.

I.H.P. per ton.				I.H.P. per square foot of grate.		Heating surface per I.H.P.		Boiler-tubes material.
Machinery complete.		Boilers.		Natural draft.	Forced draft.	Natural draft.	Forced draft.	
N. D.	F. D.	N. D.	F. D.					
						Sq. ft.	*Sq. ft.*	
7.4	11.6	12.4	19.3	10.4	16.2	2.9	1.9	Iron.
7.4	11.1	12.1	18.1	10.3	15.4	2.8	1.9	"
7.4	11.3	12.2	18.7	10.3	15.8	2.8	1.9	"
6.2	10.8	10.7	18.6	10.6	18.4	3.0	1.7	Iron.
7.3	10.7	12.6	18.5	12.5	18.2	2.6	1.7	"
6.6	10.4	11.2	17.7	10.2	16.2	2.9	1.8	"
7.3	9.6	12.7	16.8	11.3	14.8	2.6	2.0	"
7.0	11.1	12.0	19.0	10.5	16.8	2.7	1.7	"
7.1	10.7	12.4	18.6	11.2	16.7	2.6	1.7	"
6.9	10.6	12.0	18.3	11.1	16.9	2.7	1.8	"
6.7	11.6	11.7	20.4	10.8	18.9	2.9	1.7	Steel.
9.6	14.7	16.1	24.6	12.8	19.6	2.0	1.3	Iron.
10.0	15.3	16.9	26.0	11.7	17.9	2.0	1.3	"
10.4	14.0	18.1	24.4	12.2	16.4	2.0	1.5	Steel.
8.6	14.7	14.8	25.4	10.1	17.3	2.5	1.4	"
7.5	14.8	16.9	26.1	10.9	16.9	2.2	1.4	Iron.
9.2	14.7	16.5	25.3	11.5	17.6	2.2	1.4	"

TABLE D.—

Ship.	I.H.P. on trial, natural draft.	Air-pressure.	I.H.P. on trial, forced draft.	Air pressure.	Weights.		Surfaces.	
					Machinery complete.	Boilers.	Grate.	Total heating.
		Inch.		*Ins.*	*Tons.*	*Tons.*	*Sq. ft.*	*Sq. ft.*
Barrosa	2,133	0.4	3,111	1.4	248	151.3	201	4,529
Barracouta	1,920	0.9	...	...	247	148.3	195	4,529
Blanche	1,832	0.6	2,849	1.4	245	142.1	188	4,650
Blonde	1,918	0.5	2,762	1.7	245	142.3	188	4,650
Average	1 951	0.6	2,907	1.5	246	146.0	193	4,589
Kutoomba	4,538	0.5	...	...	497	288.8	377	10,150
Mildura	4,543	0.9	...	...	505	289.0	319	10,150
Wallaroo	4,574	0.5	...	...	505	289.0	376	10,150
Tauranga	4,651	0.6	...	...	541	281.5	378	9,621
Ringarooma	4,771	0.4	...	...	529	281 0	378	9,621
Average	4 615	0.7	...	...	515	285.8	365	9,938
Barham	3,618	1.1	4,561	2.1	273	138.0	244	7,088
Bellona	3,566	1 2	...	..	275	141.3	244	7,088
Average	3,592	1.1	4,561	2.1	274	139.6	244	7.088
Pallas	5 066	0.1	7,333	1.5	516	297.7	425	11,109
Pearl	5,372	0.3	7,227	1.5	543	318.4	400	11,105
Philomel	4,923	0.3	7,735	1 2	539	320 9	416	11,105
Phœbe	4,705	0.2	7,582	1.6	559	317.4	401	10,782
Average	5.016	0 2	7.469	1.4	539	313.6	410	11 025

NOTE.—The weights given in column 7 comprise boilers, funnels, casings, pipes, spare gear, and all fittings in boiler rooms, also water in boilers at working level.

THIRD CLASS CRUISERS—Continued.

I.H.P. per ton.				I.H.P. per square foot of grate.		Heating surface per I.H.P.		Boiler tubes material.
Machinery complete.		Boilers.		Natural draft.	Forced draft.	Natural draft.	Forced draft.	
N. D.	F. D.	N. D.	F. D.					
						Sq. ft.	*Sq. ft.*	
8.6	12 5	14.1	20.6	10 9	15 4	2.1	1.4	Steel.
7.8	...	13.0	...	9.8	...	2.3	...	"
7.4	11.6	12.8	20.0	9.7	15.1	2.5	1.6	"
7.8	11.2	13.5	19.4	10.1	14.7	2.4	1 6	"
7.9	11.8	13.3	20.0	10.1	15.1	2.3	1.5	..
9.1	...	15.7	...	12.0	...	2.2	...	Steel.
9.0	...	15.7	...	14.2	...	2.2	...	"
9.0	...	15.8	...	12.1	...	2.2	...	"
8.6	...	16.5	...	12.3	...	2.0	...	"
9.0	...	17.0	...	12.6	...	2.0	...	"
8.9	...	16.1	...	12 6	...	2.1	...	..
13.2	16.7	26.2	33 0	14.8	18.7	1.9	1 5	Steel.
13.0	...	25.2	...	14.6	...	1.9	...	"
13.1	16.7	25.7	33.0	14.7	18.7	1.9	1.5	..
9.8	14.2	17.0	24.7	11.9	17.2	2.1	1.5	Steel.
9.9	13 3	16 9	22.7	13.4	18.0	2.0	1.5	"
9.1	14 3	15.3	24.1	11.8	18 5	2.2	1.4	"
8.4	13.5	14.9	23.9	11.7	18.9	2 2	1.4	"
9 3	13.8	16.0	23.8	12.2	18.2	2.1	1.47	..

TABLE E.—

Ship.	I.H.P. on trial, natural draft.	Air pressure.	I.H.P. on trial, forced draft.	Air pressure.	Weights.		Surfaces.	
					Machinery complete.	Boilers.	Grate.	Total heating.
		Inch.		*Ins.*	*Tons.*	*Tons.*	*Sq. ft.*	*Sq. ft.*
Rattlesnake	...	...	2,740	2.3	133	85.2	126	4,639
Grasshopper	...	...	2,368	2.8	118	72.4	120	4,396
Sandfly	...	...	3,014	2.8	115	72.4	119	4,334
Spider	...	...	2,664	2.5	115	72.0	119	4,334
Average	...	...	2,696	2.6	120	75.5	121	4,426
Sharpshooter	2,836	0.8	...	...	167	94.3	190	5,330
Spanker	2,524	1.0	...	...	163	98.0	182	5,330
Boomerang	2,612	0.6	3,509	2.0	173	98.0	190	5,330
Karakatta	2,598	0.5	3,840	1.7	171	95.6	190	5,330
Salamander	2,825	0.9	...	...	175	109.2	153	5,319
Seagull	2,792	1.8	...	2.3	176	109.4	144	5,319
Sheldrake	2,659	0.9	...	...	...	109.0	153	5,319
Assaye	2,774	1.1	...	...	174	100.0	153	5,980
Plassy	2,895	0.9	...	...	176	97.6	153	5,980
Skipjack	2,282	0.6	3,931	3.4	170	102.5	192	5,469
Speedwell	2,601	0.5	3,588	2.8	168	102.0	191	5,469
Average	2,672	0.9	3,717	2.4	171	101.4	172	5,470
*Sharpshooter**	2,620	Nil	3,238	0.1	197	124.3	269	7,695

* After being fitted with Belleville boilers.

NOTE.—The weights given in column 7 comprise boilers, funnels, casings, pipes, spare gear, and all fittings in boiler rooms; also water in boilers at working level.

TORPEDO GUNBOATS.

I.H.P. per ton.				I.H.P. per square foot of grate.		Heating surface per I.H.P.		Boiler tubes, material.
Machinery complete.		Boilers.		Natural draft.	Forced draft.	Natural draft.	Forced draft.	
N. D.	F. D.	N. D.	F. D.					
						Sq. ft.	*Sq. ft.*	
...	...	...	...	...	...	...	...	Steel.
...	20.0	...	32.8	...	19.7	...	1.85	"
...	26.2	...	41.8	...	25.3	...	1.43	"
...	23.1	...	37.0	...	22.3	...	1.62	"
...	23.1	...	37.2	...	22.4	...	1.63	"
17.0	...	30.1	...	15.5	...	1.8	...	Steel.
15.4	...	25.7	...	13.8	...	2.1	...	"
15.1	20.2	26.6	35.8	13.7	18.4	2.0	1.5	"
15.1	22.4	27.3	40.4	13.6	20.2	2.0	1.3	"
16.1	...	25.9	...	18.4	...	1.8	...	"
15.8	...	25.6	...	19.3	...	1.9	...	"
15.1	...	24.4	...	17.3	...	1.9	...	"
15.9	...	27.7	...	18.1	...	2.1	...	"
16.4	...	29.8	...	18.9	...	2.0	...	"
13.4	23.1	22.3	38.5	11.8	20.4	2.3	1.3	"
15.4	20.7	25.5	35.1	13.6	18.7	2.0	1.5	"
15.6	21.6	26.4	37.4	15.5	19.4	2.0	1.4	"
13.3	16.4	21.1	26.1	9.8	12.0	2.9	2.3	"

TABLE E.—

	Ship.	I.H.P. on trial, natural draft.	Air pressure.	I.H.P. on trial, forced draft.	Air pressure.	Weights.		Surfaces.	
						Machinery complete.	Boilers.	Grate.	Total heating.
			Inch.		*Ins.*	*Tons.*	*Tons.*	*Sq. Ft.*	*Sq. Ft.*
F	*Alarm*	2,593	0.8	3,886	2.4	220	130.1	156	6,241
F	*Circe*	2,621	0.9	3,508	1.9	219	129.5	156	6,241
F	*Gleaner*	2,606	0.9	3,632	2 7	174	109.8	155	5,578
F	*Gossamer*	2,634	0.9	3,654	2.7	167	103.4	183	5,654
F	*Hebe*	2,702	0.7	3,566	2.0	216	129.1	163	6,241
F	*Leda*	2,696	0.8	3,597	2.2	217	128.8	156	6,241
F	*Antelope*	2,653	0.5	3,597	1.3	211	128.6	179	6,228
F	*Jaseur*	2,546	...	3.711	2.7	207	121.0	182	6,220
F	*Jason*	2,676	0.9	3,552	2.1	208	122.5	180	6,220
F	*Niger*	2,710	0.8	3,785	2.2	209	122.0	182	6,220
F	*Onyx*	2.526	0.8	3,548	2.1	210	125.1	169	6,197
F	*Renard*	2,609	0 8	3,962	2.5	210	123.6	169	6,197
G	*Dryad*	2,696	0.9	3,709	2.2	233	129.5	183	6,301
G	*Halcyon*	2,590	1.0	3,546	2.0	231	129.0	182	6,204
G	*Harrier*	...	...	...	...	...	129.0	182	6,204
G	*Hazard*	2,621	0.8	3,734	2.1	233	134.7	172	7,086
G	*Hussar*	...	...	...	...	...	129.0	...	6,204
	Average	2,631	0.8	3,665	2.2	211	124.4	171	6,204
	Speedy	3,046	0.5	4,703	1.7	212	107.2	204	17,700

NOTE.—The weights given in column 7 comprise boilers, funnels, casings, pipes, spare gear and all fittings in boiler rooms, also water in boilers at working level.

TORPEDO GUNBOATS—Continued.

I.H.P. per ton.				I.H.P. per square foot of grate.		Heating surface per I.H.P.		Boiler tubes material.
Machinery complete.		Boilers.		Natural draft.	Forced draft.	Natural draft.	Forced draft.	
N. D.	F. D.	N. D.	F. D.					
						Sq. ft.	*Sq. ft.*	
11.8	17.6	19.9	29.8	16.6	24.9	2.4	1.6	Steel.
11.9	16.0	20.3	27.1	16.8	22.4	2.3	1.7	"
14.9	20.8	23.9	33.3	16.7	23.4	2.1	1.5	"
15.7	21.8	25.5	35.4	14.3	19.9	2.1	1.5	"
12.5	16.5	20.9	27.6	16.5	21.8	2.3	1.7	"
12.4	16.5	21.0	28.1	17.2	23.0	2.3	1.7	"
12.5	17.0	20.7	28.1	14.8	20.1	2.3	1.7	"
12.3	17.9	21.0	30.6	14.0	20.3	2.4	1.7	"
12.8	17.0	21.9	29.1	14.8	19.7	2.3	1.7	"
12.9	18.1	22.2	31.0	14.3	20.7	2.2	1.6	"
12.0	16.9	20.2	28.3	14.9	20.9	2.4	1.7	"
12.4	18.8	21.2	32.2	15.4	22.2	2.3	1.5	"
11.5	15.9	20.9	28.7	14.7	20.2	2.3	1.7	"
11.2	15.3	20.0	27.4	14.2	19.4	2.3	1.7	"
...	...	...	...	...	...	...	..	"
11.2	16.0	19.5	27.8	15.2	21.7	2.7	1.9	"
...	...	...	...	...	...	...	...	"
12.4	17.3	21.2	29.5	15.3	21.4	2.35	1.7	..
14.3	22.1	28.4	43.9	14.9	23.0	5.8	3.7	Steel.

TABLE F.—COMPARISON OF WEIGHTS OF FRAMING.

Ship.	Bed plates.				Standards and guides.				Total.				Remarks.
	Tons.	*cwt.*	*qrs.*	*lbs.*	*Tons.*	*cwt.*	*qrs.*	*lbs.*	*Tons.*	*cwt.*	*qrs.*	*lbs.*	
Royal Oak....	32	8	3	12	37	6	2	16	69	15	2	0	Forged-steel front and back columns.
Ramillies.....	37	19	0	16	40	2	2	12	78	1	3	0	Cast-steel back columns; forged-steel front columns.
Crescent.....	36	1	0	24	46	0	2	10	82	1	3	6	Cast-iron back columns; forged-steel front columns.
Gibraltar.....	40	16	1	16	26	7	3	1	67	4	0	17	Forged-steel back and front columns.
Royal Arthur.	20	5	0	20	27	10	1	17	47	15	2	9	Cast-steel columns, back and front.
Endymion.....	32	0	1	12	22	2	1	22	54	2	3	6	Cast-steel back columns; forged-steel front columns.

NOTE.—The engines of all these vessels have cylinders of the same dimensions.

veloped 7,051 I.H.P. for fifty hours, with a coal consumption of 2.23 pounds per I.H.P. per hour. The *Sirius*, on a passage, maintained 4,555 I.H.P. for sixty-four hours, with a coal consumption of 2.03 pounds per I.H.P. per hour. The *Pallas*, on a passage, developed 3,620 I.H.P. for seventy-three hours.

The quarterly passage trials of twenty-four hours' duration made by these vessels furnish further indications of their ability to repeat their trial performances with natural draft. On these occasions the full natural draft power is maintained for four hours, and not less than three-fifths of the natural draft for the remaining twenty hours.

In the discussion, **Mr. Durston** called attention to the different figures given for the weight of the boilers of the *Speedy* in Table E, and in the table on page 346, and stated that the diference was due to the fact that, in one case, all the boiler room weights are included, which are not in the other.

Mr. Thornycroft in speaking of the Belleville boilers in the *Sharpshooter*, and for the *Powerful* and *Terrible*, said he thought he could show that it is too heavy. The information given in the tables was not all comparable on the same terms, because in the paper itself the boilers were compared without the weight of some of the fittings, whereas the ash-hoists and all the engine room fittings connected with the boiler were included in the tables. The boilers of the *Daring* weighed 47.7 tons instead of 48.5, which would make the I.H.P. per ton of boiler 92.5. Considering the different boilers, he found that the weight of the drum boiler with all the fittings amounted to .027 ton per square foot of heating surface, and the locomotive boiler to .020. The weight of the water-tube boilers in the *Speedy* was .007 ton per square foot. In the *Sharpshooter*, with Belleville boilers, the figure was .016.

Mr. J. P. Hall said that the weight now allowed for the machinery of second class cruisers, such as those bracket *E*, in Table C, is 800 tons, or an increase of 10 per cent.

Mr. A. E. Seaton pointed out that the engines of the *Endymion* and *St. George* had not forged steel columns, but steel castings, and were not like those of the *Royal Arthur*, but were on an entirely new design. It had been concluded that the lightest form of engine was that with turned steel columns. On inquiry he had found that a design with back cast columns could be made really lighter than with forged turned columns.

In replying to the discussion, **Mr. Durston** said that since the paper had been written other ships had been tried over long periods with the following results:

	Hrs.	*I.H.P.*	*Coal per I.H.P.*
Spartan, . . .	72	6,777	1.96
Resolution, . . .	48	8,085	2.10
Ramillies, . . .	34	8,111	1.97

The weight of water in the water tube boilers, on page 346, was for cold water, and in the Tables A to E for hot water.

TUBULOUS BOILERS IN THE FRENCH NAVY.

By Assistant Engineer John K. Robison, U. S. Navy.

In what follows, I have endeavored to place before the readers of the Journal the result of my observation of the practical working of tubulous boilers in France, with such data regarding the most important of them as could be obtained.

In nearly every vessel under construction for the French Navy the boilers are of the tubulous type, and cylindrical boilers have almost entirely disappeared from designs for men of-war in favor of the lighter tubulous ones. For torpedo boats and other vessels of small size, the boilers are generally of the Thornycroft or Normand type, and, except for slight differences in detail, are similar to those used in this country. The French seem to get better results from the Normand and Lagrafel than from the Thornycroft that were first used in their navy for torpedo boats; but this is largely due to the fact that the later boilers are of French design and manufacture.

The boilers which are designed to replace the large cylindrical ones that are used in all other countries for large vessels are those that present the most novelty to us, and are the ones which will be considered here. There are only three types which can be said to have any chance of replacing the Scotch boiler: the Belleville, the D'Allest and the Niclausse. Other types essay to fill the places of these boilers, but, so far as I know, the three named are the only types that are used in vessels larger than gunboats. Others have been tried with a view to applying them in large vessels, but so far they have not been a success.

Of the boilers mentioned, the Belleville is the oldest type; it is claimed for it that it is also the oldest tubulous boiler, and it probably is the oldest French one. It is too well known to require description, but some consideration of its working may

not be out of place, when it is considered that the results are those reached where the experience has been greater than ours.

The extremely small quantity of water in the boiler has made the use of an automatic feed regulator necessary. This regulator works well when it does work, but fails to work at all, often enough to destroy all confidence in it. Besides, when the regulator fails to work, serious accidents often result. The amount of feed water is so small that any failure of the regulator to act is liable to cause the water to disappear entirely from the boiler. The small quantity of water in the boiler likewise causes large variations in the steam pressure and necessitates a larger pressure in the boiler than at the engines; that is to say, there is a reducing valve between the boilers and the engines.

The slow circulation of the water causes the tubes to deteriorate very rapidly if the water is not pure. The tubes of the lowest row are made very thick, but they wear out very rapidly nevertheless, being bent after fires are lighted under the boilers two or three times.

The system of circulation of the water causes a great deal of "priming," and this cannot be cured even with the use of a complicated set of baffle plates in the steam drum of the boiler, and with the addition of a separator between the boilers and the reducing valve. The reducing valve must also reduce the amount of water in the steam, though, as has been seen, this is not the prime object of its use. It has been estimated by engineers that have worked these boilers for several years that the amount of water in the steam at the cylinders is never less than 10 per cent.

The use of the Belleville boilers was said at the outset to be sure to give a great gain in economy of fuel. In fact, many engineers still seem to think that they are not greatly inferior to the Scotch boilers in this respect. The fact is, however, that the arrangement for the combustion of the coal to take place entirely in one place has made the mixing of the gases of combustion very poor. To ensure their proper mixing all the gas from the grate should be brought together at some point before the combustion is supposed to be completed. This would correct the inequalities in the thickness of the fires in different

parts of the grate. So very poor is the mixing of the gases in the Belleville boiler, that it has been found necessary to have a pump for forcing jets of compressed air in the top of the furnace, forcing the gases of combustion down towards the grate, and so promoting their thorough mixing.

The absolute necessity of a sure acting feed pump has led to the use of a specially designed pump that will always be sure to act. This result is obtained at the cost of a large amount of steam for the pumps, but the result is so necessary that it has been said that the pump is what makes the boilers.

The accessories to this boiler are so numerous that they make a considerable addition to the machinery of a vessel. The number of separate machines that are required to make this boiler act in at all a safe way leads to an exaggerated amount of repairs, and the care of the steam producing plant becomes a more difficult matter than that of the engines. The repairs to the boilers are more costly than those for ordinary boilers. Not that any one case of repairs is not cheaper than a similar job would be with Scotch boilers, but the greater number of repairs has led to greater expenses for the government in the repair shops. The repairs can, however, be made in a much shorter time than with the cylindrical boilers, and this must be held to counterbalance in a large degree, the greater frequency of break downs in the boilers. Also the large number of boilers in the steam producing plant of a powerful ship makes the loss due to the putting out of commission of any one boiler a minimum. It is also to be noted that all ships that are fitted with these boilers have more boilers than are necessary to the running of the engines at full power. Thus it is always possible to run at full power even with one or more boilers disabled.

The advantages of the Belleville boiler over the Scotch boiler that are the most appreciated in France are the great gain on the weight of the steam producing plant, even with a reduction in the forcing of the boilers, and the ease of raising steam. The pressure allowed for this boiler is practically unlimited by the boiler, on account of the small diameter of the cylinders that contain the steam. The parts are small and can easily be removed from the

boiler rooms without cutting any holes in the decks. In fact, the whole boiler may be removed from the boiler room without troubling the decks at all.

The manufacturers of this boiler claim that it is possible to use salt water in it without any bad effects. Though it has never been intended that they, more than any other working at a high pressure, should be usually fed with salt water, it has been occasionally necessary to use salt water in them. The results have not been of the best. The tubes were found to be eaten away, and the rods of the feed regulator were soon covered with incrustrations that prevented it from acting, and so entirely destroyed the boiler.

As has been said, these were the first French tubulous boilers. While Mr. Belleville has been constantly at work devising methods for making his boiler run with success, other people have been busy devising some way of getting around the difficulties in the Belleville boilers by a change in the system.

The D'Allest boiler, or more properly the Lagrafel-D'Allest, is probably the best of the attempts to secure a substitute for the Scotch boilers. This boiler has been described in the JOURNAL, but some recent changes that have been made may be an excuse for going over the ground again. This description will follow, but the following is the result of observations made on the comparative working of these boilers and the Belleville. Each boiler, it will be remarked, has advantages over the other.

The D'Allest boilers are not quite so heavy as the Belleville, but the floor space occupied is greater for the same area of grate or heating surface. As, however, the D'Allest boilers are much more capable of being forced than the Belleville (which are uneconomical with over fifteen pounds of coal burned per square foot of grate), it may be said that the space occupied by the D'Allest boilers is not greater for the same power than that required by the Belleville. A great advantage for the D'Allest boilers is, that with them it is unnecessary to have more than the ordinary auxiliaries of the Scotch boilers, and the frequency of repairs with the Belleville boilers is thus avoided. The amount of water in the D'Allest boilers is not, of course, so great as in

D'Allest Boiler.

Fig. 1.

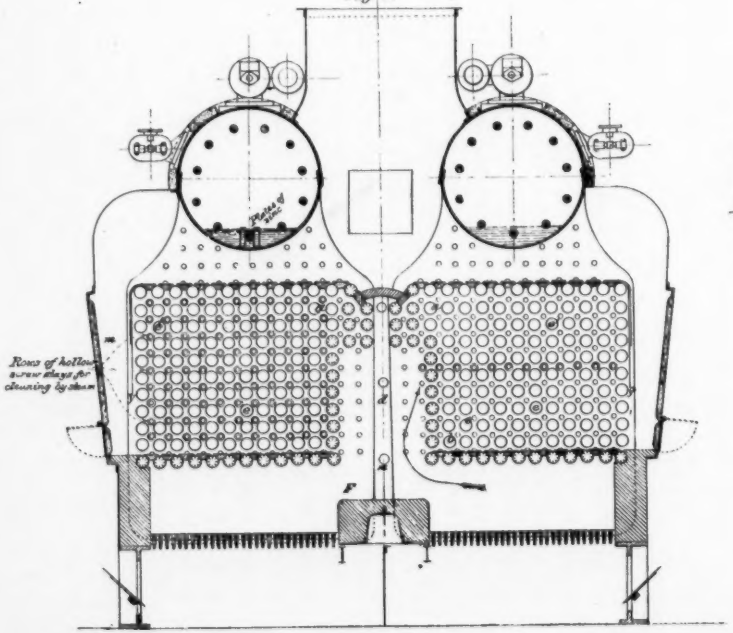
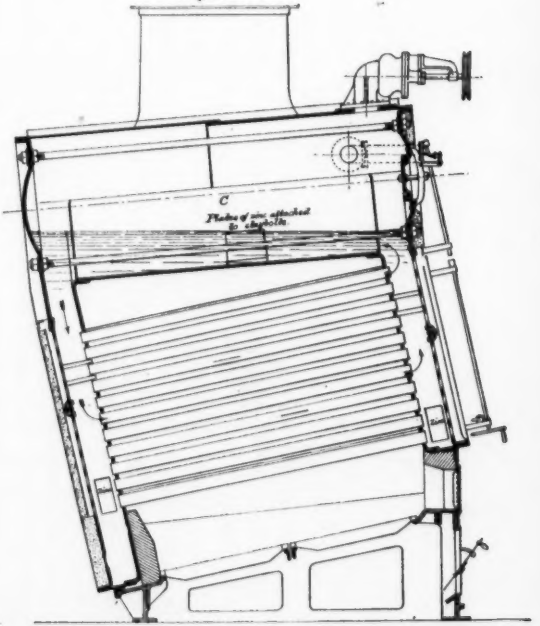


Fig. 2.



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Scotch boilers, but it is sufficient to make the use of that bugbear of the practical engineer, the automatic feed regulator, unnecessary. The cost of running the D'Allest boiler is far less than that of the Belleville, and, indeed, it may be said that on this point the D'Allest boiler may be compared with the Scotch. It has an independent combustion chamber, and thus the gases are well mixed before entering the uptake. The results of steaming with this boiler are in marked contrast with those from the use of the Belleville. While the D'Allest boilers have not required more coal than Scotch boilers for similar engines, the loss in coal has been with the Belleville boilers as much as 42 per cent., as will be seen later.

The greatest advantage of the Belleville boilers over the D'Allest lies in the comparative freedom of the tubes of the Belleville boiler to expand when heated, they being fastened at only two points in each element while those of the D'Allest are fastened, the same as the tubes of Scotch boilers, at both ends of each tube. This reduces the danger of leaky tubes in the Belleville below what it is in the D'Allest boilers. Another advantage of the Belleville over the D'Allest lies in the fact that the parts of the former are smaller than those of the latter, and that, therefore, there is less difficulty in removing them from the fire-rooms in case of injury beyond repair. When one considers the fact that the French Government requires reducing valves to be placed between the boilers and the engines whenever tubulous boilers of any type whatever are used, some excuse for the use of the Belleville in preference to the D'Allest boiler may be found. Of course, the question of the advisability of using any type of tubulous boilers is quite apart from the question of the superiority of one tubulous boiler over another.

Of the many other types that have been proposed for replacing the Scotch boilers there is but one, the Niclausse, that has so far been recognized as possessing the points that are requisite for use in men-of-war. These boilers are modifications of the Collet, that have been fully described in the JOURNAL. The differences between the new boiler and the older one lie almost entirely in the details of construction, the main points of the boilers

being the same. These boilers are as different from the others as the latter are from each other. They are compared only with the D'Allest, as the latter are so evidently superior to the Belleville that it would be waste of time to include a second comparison.

The weights of the D'Allest and the Niclausse boilers are practically the same for the same area of heating surface, with the same advantage for the Niclausse in regard to the space occupied as for the Belleville boiler. But while the Niclausse boilers are capable of being forced more nearly to the power of the D'Allest than the Belleville boilers are, they are not the equals of the D'Allest in their capacity for high powers. It may, therefore, again be said that the D'Allest boiler takes up less space for the same power than the Niclausse. Both of these boilers give dry steam at the highest powers at which they are run, and, therefore, have a point of advantage over the Belleville. The amount of water in the Niclausse is less than in the D'Allest, but it is still large enough so that the water level may be easily maintained without the use of any other than the ordinary check valves on the boilers. The greater the amount of water in any boiler, however, the better it is for keeping a steady steam pressure; and some difficulty was experienced in maintaining the pressure of steam constant during the forced draft trials of the *Friant* (fitted with Niclausse boilers). The frequency of repairs to one of these boilers is about the same as for the other, and the cost is about the same in each case. The joints in the Niclausse boiler are all metallic and conical, and so require more care in the making, but are less liable to give trouble when once made. The tubes are all free at one end, and therefore the danger of leaky tubes is reduced to a minimum. In fact, during all the trials of the *Friant*, there were no leaks in this boiler. It is another point in its favor that the repairs are all made from the front of the boiler. It must also be remarked that it is easier to mount and dismount a tube in the Niclausse than in any other type of tubulous or other boiler. The complete operation of removing a tube and replacing it with another took, in one instance within my observation, less than two minutes.

The advantages that the D'Allest boilers have are chiefly in the matter of economy. As has been said, they are about as good as the Scotch boilers, while the Niclausse or the Belleville give much poorer results in actual use than have been found from the use of ordinary boilers. Another advantage of the D'Allest boiler lies in the fact that the tubes in the rows next to the fires are all Serve tubes, and thus much less liable to burn out than ordinary tubes. It would be hard to use this type of tubes in the Niclausse boilers, on account of the inner circulating tube in each element. It would seem that tubes of the style used in the Niclausse boiler are the best, however, on account of their freedom to expand when the boiler is being fired. The one great advantage of the D'Allest boilers seems to lie in their great relative economy over any other tubulous boilers. Mr. D'Allest himself says that this advantage is almost if not entirely due to the use of an independent combustion chamber. There would be little difficulty in adding a combustion chamber to the Niclausse boiler, and then it would seem that this boiler would be inferior to the D'Allest in but the detail of the amount of water in the boiler. This defect could be remedied by a change in the size of the tubes to allow for the increased rate of evaporation rendered possible by the addition of the combustion chamber, and by the use of a larger steam drum at the top of the boiler. Perhaps even now it may be said to be a question whether the Niclausse boilers are not the equals of the D'Allest, but the opinion of French engineers seems to be that the D'Allest are the boilers of the future, and that, with a few changes, they can be readily supplied in the place of Scotch boilers. The addition of hydrokineters would reduce the disadvantages of the D'Allest tubes being fixed at both ends. This apparatus has not yet been used in these boilers.

In the use of these boilers in the French Navy it is to be remarked that even with the number of spare boilers (20 per cent. in many cases), and with the small rate of combustion allowed in all cases (never above 31 pounds of coal per square foot of grate), and with engines that are considerably heavier than those used for the same power in this country, the total weight of the

machinery is not so great as in our latest ships. In no case of a modern French man-of-war fitted with tubulous boilers, that I now recollect, has the weight been over 200 pounds per I.H.P. of all the machinery; in most cases the weight is down to about 185 pounds. These figures are, of course, for large vessels of the battle-ship or fast cruiser type. This seems to be the greatest if not the only advantage for the tubulous boilers. The pseudo advantage of quickness in raising steam is one that is more than counterbalanced by the always attendant greater difficulty in managing the boilers when under pressure.

The pertinent points that seem to me to need attention in the French boilers are that the tubes are always so arranged so as to be easily removed or cleaned. This seems to be an absolute requisite for any boiler that can entirely replace the Scotch boilers.

Tubulous boilers will always give more trouble to keep in good condition than would Scotch boilers, but they are sure to retain their full efficiency almost indefinitely, as the worn parts are replaced by new ones that are as strong as the old ones were in the first place. There is no shell to deteriorate.

It is also to be remarked that the tubes used in these boilers are invariably of a larger diameter than is generally used in the boilers made in this country. The gain in the weight of the boilers may be said to be about equal to the weight of the water in Scotch boilers that would have to be substituted for the tubulous boilers.

THE BELLEVILLE BOILERS OF THE *AUSTRALIEN*.

The Messageries Maritimes is one of the greatest steamship companies of France, if not the very greatest. This company began the use of Belleville boilers some five or six years ago. The officials of the company seem to be fairly well satisfied with the performance of them, but there is a tendency to obtain something that will give better economical results. Designs were prepared in the last vessel designed (the *Ernest Simon*) for the use of Scotch boilers. There was still a question as to which type of boilers would be adopted when I left France.

Of the vessels that are fitted with Belleville boilers in the fleet of this company the *Australien* is the oldest. She is a vessel about 466 feet in length, and of about 10,000 tons displacement. The mean speed on the trial trip was 17.5 knots. She has a single screw vertical engine of 7,535 I.H.P. on the trials, and developing about 4,100 I.H.P. on the whole trip to Australia, giving a mean speed of about 15 knots. The engine was constructed in 1890, at the company's shops at La Ciotat, and is a triple expansion engine with three cylinders. The evaporating plant is composed of twenty Belleville boilers for the main engines, and of one small cylindrical boiler for the distillers and for the winches on deck. These boilers (Belleville) have five, or for the most part six, elements each, and are twelve rows high. The total grate surface is 581.27 square feet.

The weight of the machinery is as follows:

Engines, tons	750
Boilers and accessories, tons	392
Water in boilers, tons	20
Total for machinery, tons	1,162
Per I H.P. (on 7,535 I.H.P.), pounds	340
Engines per I.H.P., pounds	220
Boilers per I.H.P., pounds	115
with water per I.H.P., pounds	120

The fire room is fore-and-aft.

The watch for four hours consists of one engineer, two chief firemen, three first-class firemen, two Arab leading men of the fire room, eight Arab firemen, six Arab coal passers, two European oilers, two Arab oilers; total, twenty-six men.

There have been some more or less important changes made in the fire room from the original designs. The pipe connecting the feed regulator to the boiler went originally directly to the ejector. It was found that this pipe soon filled with sediment, and caused the feed water to fail in the boiler. The pipe was changed to go directly to the feed-water collector at the base of the boiler. A tube was burned out in finding that this change was necessary.

The lever of each feed regulator has been fitted with a wire

handle, so that it may be seen at any time whether the regulator is in working order or not. This is done by opening the regulator wide, and seeing if the valve closes when the handle is loosed.

RUNNING OF THE BOILERS OF THE *AUSTRALIEN*.

Filling with water.—The boilers are always filled with fresh water, and never has any other water been used for this purpose. In the instructions sent to the company by Mr. Belleville, provision is made for the use of salt water for filling the boilers, but the accidental use of salt water in making up feed gave such bad results that the company has never used it for originally filling the boilers.

Raising steam.—The grates are carefully cleaned and put in place. They are entirely covered with coal to a thickness of about four or five inches. At the front of the boiler a wall is built up around the furnace door. This wall of coal and kindling wood is lighted first. Before lighting the fires the safety-valves are opened. The ash-pit doors are closed, and the furnace doors are left slightly open at first. The doors to the tube nest are closed.

When the little wall around the furnace door is well lighted, it is spread over the grate and the furnace doors closed; the ash-pit doors are then opened more or less according to the time allowed to raise steam. This time is generally about 1½ to 2 hours. Steam can be raised in less time, and it has been done in ¾ hour, but the effects on the boiler are not good, and the rules of the company call for 2 hours in raising steam.

When the elbow communicating between the ejector and the collector on the boiler begins to warm up, the blow-off cocks are opened several times, and the boiler freed of the sediment in the base of the tubes. This serves to promote circulation in the boiler during the time that there is no steam drawn off to make the circulation sure. If this is not done, the water hammering in the tubes during the time that steam is being raised will cause

trouble and may break any or all joints in the tubing of the boiler.

The feed pumps are kept going from the time the fires are lighted. As soon as the pressure of steam in the main boilers becomes great enough to run these pumps, the steam is taken from the main boilers. (Pressure as much as 60 pounds.)

Getting Under Way.—If there is any delay in getting under way after steam is raised, the ash-pit doors are closed and the furnace doors opened. The feed pumps are kept working as always while the fires are lighted, and the extra steam is condensed. If the fires are properly managed there is no danger of steam blowing off from the safety valves, and it is rarely necessary to send any steam to the condenser.

The stop valves are not opened wide, nor is the pressure in the boilers allowed to exceed 150 pounds, until the engines are well under way.

When the engines are first started, there is great danger of "priming." The water level must be carefully watched and the feed regulators kept closed at this time. The fires are not forced until after the engines are started, nor are the doors opened.

The doors of the tube nest are never opened except to clean tubes.

Working of the Fires.—The firing must be thoroughly regular and methodical. There has been great difficulty in finding European firemen to do the work on these boilers. It is always necessary that the fires should be what would, in Scotch boilers, be considered low, that is to say, they are never more than five or six inches thick. At the same time it is absolutely necessary that there should be no holes in the grate, or the air would rush through at this point alone and leave the gases of combustion incompletely burned. The firemen that have been accustomed to the grates of the Scotch boilers, seemingly cannot learn to properly fire the furnaces of the Belleville boilers, which are so much wider than those of the Scotch boilers.

The grates are fired in rotation, beginning forward and work-

ing aft, and putting two or three shovelfuls of coal on the fires at a time. The tendency of the firemen is to fire only on the front of the grates, and care must be taken to prevent this: one grate is always in process of being fired. When those in one fire room have been fired, those of the other fire room are commenced, and so on until all the fires have been fed, when the operation is recommenced.

The safety valves are set at 220 pounds, and the steam at the cylinder is only at 185 pounds pressure. Variations in pressure between those limits have then no effect on the engines; but the limited quantity of water in the boilers makes rapid change of pressure frequent. A man is always at the gauge watching the pressure. The moment the pointer commences to drop the fires are pushed, and again the moment the pressure commences to rise the ash-pit doors must be closed. Failure to observe these precautions may cause the pressure to drop in one or two minutes from 200 pounds to 150 pounds, or it may increase to 250 pounds in as little time. The instant the pressure commences to rise is the one taken for cleaning fires.

Each grate is half cleaned every four hours, making the total time between cleaning fires eight hours, and preventing any part of any grate from ever getting very dirty. Two grates are often cleaned at a time, and the twenty grates are cleaned during each watch (half).

The cleaning of these grates is done in a slightly different way from that used with the grates of Scotch boilers. The part that is to be cleaned is cleared of coal, the burning coal being pushed on the other half of the grate. The clean part of the grate is then covered with fresh coal, and the clinkers, &c., on the dirty part of the grate are removed as soon as possible. The coal on the clean part of the grate is then spread over the whole grate, and the doors fixed as before cleaning fires. While cleaning fires, the valve on the feed pipe to this boiler is partly closed to prevent the boiler from filling with water. The mixer of the gases of combustion is left in operation during the cleaning. If care is taken in choosing the moment for cleaning fires the steam pressure will not fall during the operation.

The mixer of gases admits the air to the furnaces at a pressure about 5 pounds above that of the atmosphere. The air enters the furnaces through little nozzles at the top of the front of the boilers and is so directed as to force the gases of combustion downwards toward the grate. This is calculated to mix the gases and so to ensure the thorough burning of the coal. It is to be noted that the boilers were worked during an entire trip without using the gas mixer, and without any appreciable difference in the amount of coal used.

Feed.—All make-up of waste feed is with fresh water. A distilling plant is placed on the ship for this purpose. The steam is formed in the donkey boiler and is sent to the second receiver on the main engines. A connection is fitted from the donkey boiler to the exhaust pipe which permits the direct condensation of the steam in distillers such as are used in our Navy.

Lime is added to the feed water in a continuous fashion by means of a lime tank fitted above the air pumps and having an exhaust to the bottom of the condenser. This lime tank is fed by a small pipe from the feed pump discharge. Lime is added at the commencement of a voyage in the proportion of one pound for each pound of oil used in the cylinders. At the end of the voyage the ratio between the amount of lime and oil is increased to three.

The feed pumps are so arranged that any one of them can feed any boiler from either of the feed tanks. On the pipe to each boiler there is fitted at the main feed pipe a valve, and there is another graduated valve at the entrance of the feed water to the feed regulator. This graduated valve is much used in running the boiler, for the feed regulator is not entirely depended upon. The graduated valve is kept as far closed as possible, and allows the feed to enter the boiler in an amount slightly exceeding that required for the boiler. The regulator, which is almost always nearly wide open, is thus unable to let much more water in the boiler than is absolutely necessary. During rough weather the graduated valve is more finely fixed still, so that the feed regulator cannot let too much water into the boiler, and so that it is necessary at intervals to open the graduated valve to give

the boiler enough water. In some, this graduated valve takes the place of the regular check valve on Scotch boilers, and the feed regulator is only depended on to do the regulating of the feed more finely than would be possible by hand. In this way only can any dependence be placed on the feed regulator. The check valve of the boiler is included in the graduated valve and not in the feed regulator.

On leaving the graduated valve the water enters the feed regulator, and from there enters the cylindrical reservoir at the top of the boiler. After passing the entire length of the reservoir, the water enters the ejector. It would seem that the water is heated sufficiently by its passage through the reservoir to deposit the greater part of the solid matter it contained on entering the boiler. The ejector keeps the greater part of this matter, and it may be removed from the boiler by use of the blow-off valve on the ejector. This blowing down is sometimes done in a continual fashion by opening a cock on the ejector and leaving it partly open during all the time the boiler is in use. The steam on leaving the ejector passes to the feed water collector on the bottom of the elements, and from there to the tubes. The water level in the boiler is about at the middle of the height of the tubes.

The ejector is used every 12 hours on each boiler. The valve is simply opened wide and then closed. This is repeated two or three times, and, as already said, may be used in a continuous way. This latter method is not used except in the case of leaky condenser tubes and the consequent presence of salt water in the boiler. Towards the end of a trip, the blow off cock is used every four hours.

With salt water in the boiler for any cause, the fires are forced and the circulation is thus kept rapid. Lime is added to the feed water in sufficient quantities to neutralize it completely, and frequent use is made of the blow-off cocks. The saturation of the water should not be allowed to exceed 2.

The amount of lime added to the feed is about eleven pounds per ton of feed water. The throttle valve should be closed as much as possible, and thus the "priming" is reduced. There is more danger of "priming" with salt water than with fresh water,

and as even with fresh water there is never less than 10 per cent. of water in the steam at the cylinders, there is great danger in the use of salt water of getting so much water that the working of the engines would be interfered with. There is always danger of tubes giving way with the use of salt water, and the boiler is then of no use until the tube is replaced. This is of little danger in any other way, as the breaking of a tube is dangerous to those in the fire room only if the furnace door is opened by the first shock. After that there is no danger of the steam entering the fire room. The average life of the tubes is from two to three years. The causes of burning are the failing of the feed, or the internal corrosion in the tubes, which is the result of use in whatever conditions.

The feed regulator is liable to fail in use. The wire test placed on each regulator is used as often as possible, say every hour. Nevertheless, the regulators are apt to give trouble, by "going to sleep." This may be caused by the formation of deposits on the regulator rods, or on the valve itself. Even dust on the outside of the regulator, on the moving parts, is liable to cause it to fail to act. As one of the engineers on the *Australien* said: "A regulator may not fail for four hours, but then again three or four of them may fail in that time." All the water levels are protected, and fully as much attention is paid to them as with the ordinary boilers, but frequently the regulator will fail, and the drop in the water of the boiler will not be noticed till too late. The great trouble with the regulator is that when the water is lowest the effort on the regulator is the greatest, and several times they have opened after the water had entirely disappeared from the boiler, and even when the boiler was red hot. In that case it is clear something had to break. It was always a tube or tubes. In one case, tubes of every element in a boiler burned out at the same time, and due to the above cause. This failure of the feed, due to a failure of the automatic feed, is the cause of most of the accidents to the boiler. The engineers of the Messageries Maritimes have tried to do without the regulators, but have not succeeded in regulating the water well enough in this way to run them without a man on each boiler

with that for his duty and nothing else. As soon as the regulator is found to have failed, which will generally be discovered when a tube bursts, the fires are drawn from that boiler. It requires about ten minutes to empty a boiler and to burn out a tube after the regulator fails.

Steam.—The baffle plates in the reservoir at the top of the boilers do not prevent the carrying away from the boiler of about fifty per cent. of water in the steam. The separator reduces this amount about five fold, and the reducing valve makes a further though unimportant reduction in the amount of water in the steam. There is not, in general, enough water in the steam to interfere with the working of the engines, though this sometimes occurs. The trap on the separator does not work well. It is designed to be automatic, but is always worked by hand.

The reducing valve works well and maintains a constant pressure at the engines while the pressure at the boilers is above 185 pounds. When the engines are working slowly, or when they are stopped, the pressure at the engines has a tendency to be the same as that at the boilers, but as the pressure at the boilers is then generally below the working pressure of the engines, there is no harm in this inconvenience, common to all reducing valves.

Feed Pumps.—It is said that these pumps are the soul of the boilers. When there is too much water in the boilers the pumps stop, and on the contrary if the water in the boilers gets low the pumps go very fast. This seems to me to be the result of the more or less complete action of the feed regulators. The pumps are, however, of the design that is necessary for these boilers, with large steam cylinders and consequent sure action of the pumps. The action of the boilers depends entirely on the action of the feed pumps, and these pumps must be sure in their action. The ordinary feed pumps that were used in part on these boats have not given satisfaction, and the Belleville pumps are the only French feed pumps that can be used.

Sweeping the Tubes by Steam.—At first, the boilers of this ship were cleaned as is usual with all boilers, by steam. No great difficulty was experienced with the arrangements for sweeping the tubes, and the satisfaction was perfect in all but one respect.

The quantity of steam required was so great that it was not possible to make enough fresh water to make up the waste feed. After the trial of salt water for this purpose, the bad results on the boiler tubes were so apparent that it was seen that only fresh water could be used. For that reason, the sweeping of the tubes with steam has been entirely discontinued.

Now, the practice is to have three or four of the twenty boilers out of use at all times, so that all may be cleaned in rotation, and each one after about three days' use. As soon as the boiler is cold it is cleaned, fires lighted, and another boiler put through the same treatment. The tubes of the lowest row are often covered with silicious barnacles, which are carefully removed every time the boiler is cleaned.

Banking Fires.—This is done in the ordinary way. The pressure in the boilers is reduced to about 75 pounds, the blow-off cocks frequently used, and the feed pumps kept running slowly. The blow cocks are used as when under way. Banking fires in these boilers is bad for the tubes. The water has no circulation to speak of, and that in the lower tubes soon becomes highly concentrated. The slow evaporation sends the steam to the reservoir at the top of the boiler and the steam being there condensed the lightest water remains at the top of the boiler, and the heavy water, containing all the sediment, at the bottom and in the lowest tubes.

Cleaning the Boilers Completely.—The outside of the boiler is cleaned by brushes in the regular way. The inside is cleaned after being washed with caustic soda. For this washing, about 35 pounds of caustic soda is put in each boiler. The boiler is then completely filled by adding fresh water. The fires are then lighted and a pressure of about 15 pounds is kept up for two or three hours. Then the feed water collector at the bottom of the elements is opened, and the height of the water reduced to the normal working level. The fires are then worked and the engine turned at the dock. This is kept up for three or four hours. The working of the engines cleans not only the boiler but also the piping and the cylinders, the amount of water carried over to the engines by the steam being sufficient for this. The boilers are

then completely blown down and the water put overboard. It will be found that the water contains a great deal of oil, and even that there are solid particles in the water thus thrown away. This cleaning is done every three months.

After the boiler has been washed, the joints of all tubes are broken, and the mud that remains in them is removed while it is still soft. The quantity of this mud is not great. During this cleaning of the inside of the boiler, care is taken to look out for corrosion, or the beginning of the burning of a tube.

The washing of the boilers is done while they are still warm from working, but after the fires have been drawn, of course.

As summed up by one of the ship's engineers:

"The great advantage of these boilers is the ease with which they can be repaired. A tube can be replaced in two hours, and this is the most frequent accident to the boilers. The tubes are the weakest part.

"The Belleville feed pump is what makes the boiler. It is perfect.

"The ejector is good for what it is intended for. With fresh feed water, even without a filter (which would be a great improvement), and by adding lime to the feed all the time, there is no danger from any deposits in the tubes. There will always be some deposits in the tubes, but nothing to speak of.

"The reducing valve is very good, and always works well.

"It is rare that there is sufficient priming to interfere with the working of the engines, although it is always considerable.

"It is not possible to keep up the water level by hand, except by using a man for each boiler for this purpose. The great trouble comes from the use of even a small quantity of salt water in the boilers. The regulator is soon clogged up and does not work, and then firing must be stopped. Yet some feed regulator is indispensable.

"The firing is difficult. An ordinary fireman can never succeed here.

"The Belleville boilers are much less economical than the ordinary ones, and the cost of making repairs to them is greater than for Scotch boilers. Besides this, their first cost is greater.

"Great care has to be exercised with all these delicate machines, and you can never tell what will be the next thing to break down.

"If you want Belleville boilers you must have engineers everywhere.

"It seems to me that there is no doubt that the ordinary (Scotch) boilers are the boilers to have, and that there is no use of bothering one's self with all this machinery.

"If the regulator sticks, the boiler must be put out of use.

"The automatic separator trap is not satisfactory, and is always worked by hand."

An attachment has been fitted permitting the fires to be drowned with water while on the grates.

The Belleville boilers show a saving in weight over Scotch boilers about equal to the weight of the water in the latter, and a saving in space of about 7 per cent. This saving in space varies according to the ship from zero to 10 per cent.

The ratio between the heating and the grate surfaces is about 30.

The cost of running the engines of the *Australien* during the year 1893 was 2.30 pounds per I.H.P.

The estimates for this ship called for a speed of 19 knots on the trials (17.52 realized), and for 17 knots in ordinary running (14.60 realized). The cost of the power was set at 1.54 pounds per I.H.P. It is evident that these boilers have not been remarkable in their economy.

On the ships of this line which use Scotch boilers, and those of an old type with pressures of 90 and 100 pounds, with old compound engines, the cost of a horse-power during the year 1893 was 2.02 pounds per hour. If the use of triple expansion engines gives an economy of 20 per cent., the cost of the Belleville boilers in coal alone exceeds that of Scotch boilers by 42 per cent.

If the *Australien* had been fitted with Scotch boilers she could have made a trip to Australia with a less weight of coal and machinery than with the present boilers. The question of cost, it will be remembered, has not been touched upon.

BELLEVILLE BOILERS IN THE FRENCH NAVY.

Nearly one half of the vessels now being constructed for the French Navy are to be fitted with Belleville boilers. Those now in use have given fair satisfaction.

On the ships now fitted with these boilers, a filter is always used between the air pump and the feed pumps. Lime is always added to the feed water in much the same way as on the *Australien*. The automatic feed regulators have given poor results in the Navy, and the satisfaction is less than with the Messageries Maritimes. In some cases the feed regulators have been taken off the boilers, and the feed regulated entirely by hand. Great difficulty is experienced in properly regulating the check valves, but this is thought to be better than relying on the regulators.

The lack of economy of these boilers has been condemned in the Navy as in the merchant marine. On the trials of the *Brennus*, a battleship fitted with Belleville boilers, and the largest ship, in point of power at least, to be fitted with them, the results of the preliminary trials were most unsatisfactory. The coal per I.H.P. was 3 95 pounds on one trial, and later, when the firemen had become more accustomed to the boilers, this figure was reduced to about 2.45 pounds. This is at the most economical rate of speed for the ship. In calculating the consumption of the engines, there is an auxiliary boiler that supplies all the auxiliary machinery except the air and circulating pumps, and the power of the main engines is used for the calculations. The power of the feed pumps is neglected and acts as a loss for the boiler furnishers. Neither of these trials was long enough to necessitate the fires being cleaned.

On the official trials I suppose these figures for the cost per I.H.P. were much improved. It may be interesting to compare these trials with those of the boilers of the *Jemmapes* fitted with D'Allest boilers, and with those of the *Friant* fitted with Niclausse boilers. The *Jemmapes* burned at the reduced rate of speed for which the Belleville boilers in the *Brennus* were tried, 1.84 pounds, while the *Friant* burned 1.94 pounds at the same reduced power. These trials took place at the same time and at the same place (Brest). The engines did not come from the same

place, and were not of the same type. Those of the *Brennus* were probably the best, though those of the *Friant* were about as good. The engines of the *Jemmapes* were as good as those of either of the other ships, except that they were horizontal instead of vertical, as the others were.

THE D'ALLEST BOILERS OF THE *CARNOT*.

Dimensions :	
Pressure of steam at boilers, pounds	213.6
Grate surface (of 8 of the 24 boilers on the ship), square feet	361.73
Heating surface of the tubes, square feet	10.316
surface of the water legs, square feet	404.8
Total heating surface of 8 boilers, square feet	10,720.8
Number of tubes	1,592
Interior diameter of tubes, inches	2.874
Exterior diameter of tubes, inches	3 15
Length of tubes between sheets, feet	7.88
Section of opening of front ash pit doors, square feet	9.47
above the middle brick wall, square feet	9 04
of opening between tubes at entry of gases, square feet	7.75
of opening between tubes at discharge of gases, square feet	7
of smoke pipe for 8 boilers, square feet	61.9
Volume of steam in boilers and steam collectors, cubic feet	1,059.5
Weight of water in boilers, tons	24.8
Weight of one boiler :	
Front water leg, outside sheet, pounds	735
tube sheet, pounds	880
Back water leg, outside sheet, pounds	1,129
tube sheet, pounds	1,124
Forward end of steam drum, pounds	440
After end of steam drum, pounds	440
Angle iron on forward water leg, pounds	264
after water leg, pounds	374
Shell of steam drum, pounds	3,080
Side of forward water leg, pounds	627
after water leg, pounds	561
Rivets (for the heads only), pounds	330
One manhole plate complete, pounds	304
129 tubes at 30.8 pounds each, pounds	6,129
398 tube plugs complete, at 2.2 pounds each, pounds	876
16 screw stays for the steam drum, pounds	1,197
Screw stays for the water legs, pounds	1,155
Angle irons for attaching the smoke pipe, pounds	334
Reinforcements for the safety valves, pounds	77
Total weight of the boiler without water or fittings, pounds	20,056

MOUNTING OF THE D'ALLEST BOILERS OF THE *CARNOT*.

The first D'Allest boilers to be manufactured at the government shops at Indret were those of the *Carnot*. The design came from the inventor, who sent with the drawings a set of instructions for the mounting of the boilers. Those instructions are as follows:

Tube Plates.—After the forging, chamfering and drawing out of the edges of these plates is finished, the holes for the tubes and those for the water-leg stays are drilled in the tube sheet. The holes for the tubes are 3.15 inches for the rear tube sheet, and 3.23 inches for the forward tube sheet. The holes for the screw stays must be carefully centered between the holes for the tubes. After this operation the tube sheet is assembled with the angle iron attached to it in the design.

Plug Sheet.—These plates are assembled with the ends of the steam drum as soon as they have been forged, chamfered and drawn. The holes for neither the tubes nor the stays are made at this time.

Assembling the Water Leg.—Each tube sheet is then assembled with the corresponding plug sheet. Then the side of the water leg so formed is fitted to the plates. When the joints are faired the rivet seams *a*, *b*, *c*, *d* and *e* are made.

Assembling the Water Legs Together.—Each tube sheet having been united with the corresponding plug sheet, and a water leg having been thus formed, the two water legs of a boiler are then placed at the designed distance, and the sheets of the steam drum mounted after being faired. When the joints between the drum and the water legs are fair the holes for the rivets of the seams *ab*, *cd*, *ef* are drilled, and the location of the various accessories on the steam drum fixed. The tube holes in the tube sheets are then projected on the plug sheets. The holes in the plug sheets must be exactly opposite those in the tube sheets.

It is a good idea to fix the water legs at a fixed distance from each other as soon as the boilers are assembled, and before the rivet holes are drilled or the tube holes on the plug sheet marked out. For this purpose a brace *G* is fitted between the water legs.

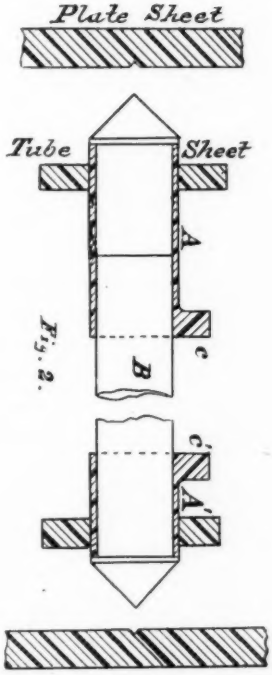


Fig. 2.

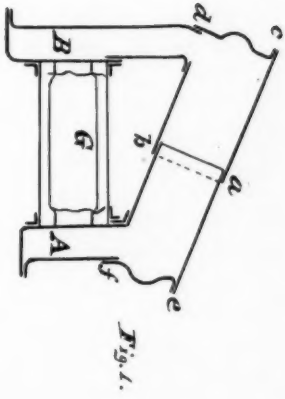


Fig. 1.

To project the holes on the plug sheet from the tube sheet some such contrivance as that shown in Fig. 2 must be used.

A, *A'*, are hollow mandrels passing in the holes in the tube sheets with an easy friction and sliding on a mandrel *B*. At the ends of the mandrels *A*, *A'*, are cones of hard steel, forming points. To project the holes from the tube sheets to the plug sheets it is necessary simply to hammer on the projections *C*, *C'*.

Drilling the Holes in the Plug Sheets.—The whole boiler is then dismounted, and the holes in the plug sheets are drilled. The plug holes are all 3.27 inches in diameter for the back, as well as for the front tube sheets. After the holes are drilled in the plug sheets, the surface around the outside of the holes is faired so as to give a good bearing surface for the rings used in forming the joints of the plugs. The faired diameter is about 4.15 inches.

The sheets are then sent to the annealing furnace and are carefully annealed.

Riveting.—After annealing, the plates are reassembled and riveted. A hydraulic riveter is used with a pressure of 80 to 100 tons on the stamp. All the seams are caulked. As many as possible are caulked on both sides.

Screwing and Placing of the Stays.—The screw stays are threaded and put in place by the ordinary method. They are riveted at both ends.

First Trial of the Boiler.—When the boiler is riveted, caulked, and the stays riveted, it is tested for the first time before the tubes are put in place. This is done because it is advisable to be sure that all the seams are tight before the tubes are in place, so that any change may be made easily. For the first trial, the plug sheets are closed by the regular plugs. The holes for the tubes in the tube sheets are closed by plugs of the same form as those used for the plug sheets, except that they are of a smaller diameter. These plugs are used in succession for all the boilers. The pressure for this trial is about 327 pounds.

Putting the Tubes in Place.—The tubes are then put in place, and expanded in the usual way. There are no stay tubes in the boiler, and all the tubes except those of the bottom row are ordinary tubes. The tubes of the bottom row are Serve tubes. It

is best to expand the tubes with wire between the expander and the tubes.

Second Trial of the Boiler.—After the tubes are in place the boilers are tested as before by hydraulic pressure. This trial is to see that there are no leaks at the joints of the tubes and to correct any other leaks not corrected after the first trial.

Mounting of the Boiler on its Supports; Mounting the Uptakes.—The boilers, assembled two by two, are then mounted on their supports, and the uptakes and smoke pipes mounted. There must be no leaks in the joints of the uptakes or the draft of the boiler will be ruined. To prevent these leaks asbestos is placed between the plates.

Steam Trial.—When the boilers are completed they are tried for coal consumption at different rates of combustion. They are then sent to the ship to be mounted on board.

Modifications.—The above process was modified with the boilers of the *Duchayla* by securing the fair bearing of all plates of the boilers before drilling any holes for the tubes either in the tube sheets or the plug sheets.

COMPARATIVE TESTS OF NEW AND OLD STYLE D'ALLEST BOILERS.

The following is the description of several tests made by the French Government of a D'Allest boiler. The description is by Mr. D'Allest. The tests determined the use of these boilers in the Navy.

Description of the Boiler.—The D'Allest boiler is modeled as far as the circulation of the gases is concerned after the Scotch boiler. This was with the hope of increasing the small efficiency of the tubulous boiler so as to have it equal to that of the Scotch. The aim has been to give the gases a chance to burn after they have been effectively mixed. Such is the function of the combustion chamber.

The distance between two tubes multiplied by the length of a tube will give a section corresponding to the tube section in the ordinary boilers.

When the gases leave the nest of tubes they are at a comparatively low temperature, and so occupy a smaller volume. In

this boiler, these gases would tend to circulate at the top of the vertical row of tubes furthest from the combustion chamber and to leave the tubes at the bottom of this row of tubes without heat. To avoide this a baffle plate is placed at the top of this row, forcing the gases to leave the tubes at the bottom of the side furthest from the combustion chamber.

The ratio of the heating to the grate surface is in the design of these boilers about 30; in those of the *Carnot* it was 29.64.

The space occupied by these boilers is as follows:

Height, 13 feet 1½ inches; width, 14 feet 5.2 inches; depth, 10 feet 2.1 inches. That is to say, the volume occupied is 0.716 cubic feet per square foot of heating surface.

The working pressure of these boilers is 213.6 pounds, and the weights are as follows:

Total heating surface of one boiler, square feet	1,340
Weight of boiler without water or fittings, pounds	20,056
Weight of supports, doors and grates, pounds	5,830
Weight of water in boiler, pounds	6,850

This is equivalent to 639 pounds per square foot of grate surface, or 21.6 pounds per square foot of heating surface of the boiler without water; and as the weight with water and fittings, 789 pounds per square foot of grate, or 26.7 pounds per square foot of heating surface.

Method of Making the Trials.—The quantity sought in each of these trials was the evaporative value of the coal in the boiler. The amount of coal burned was increased from 10 to 31 pounds per square foot of grate per hour.

The coal was weighed, the water pumped in the boiler was weighed, and the chimney gases were carefully analyzed. The temperature of the gases was taken at several places in the boiler, and the amount of priming shown by the calorimeter in each trial was found to be negligible.

The boilers used were old style Lagrafel changed for the trials into the new form. The pressure in the boilers was kept down to about 50 pounds, so as not to endanger the old boiler.

To obtain a certain fixed rate of firing of the grates, the quan-

tity of coal to be used in each charge was weighed in advance. The furnaces of each boiler were fired in turn, and at regular intervals of ten minutes. The charges of the two boilers coupled on the same combustion chamber were five minutes apart.

The coal used in all the trials was Cardiff-Cory. The ashes were weighed after each trial, and were found to be about 8 to, 10 per cent.

The measuring of the feed water was done at the feed tanks which were carefully graduated.

Gases were taken from several points in the travel of the gases: under the top of the furnace, above the brick wall in the combustion chamber, and at different points in the nest of tubes. These gases were taken at the same instant, so as to be sure of the circulation of the gases of combustion. The instant chosen was that of firing the grates.

From the trials the volume of air necessary for the combustion of one pound of coal was found to be 15.2 cubic feet.

The temperature of the gases was taken at different points: in the combustion chamber at *d*, in the uptake at *m*, and at the points in the nests of tubes indicated by *a*, *b*, *e*, *e'*. The temperature was measured by means of small bars of copper. These were plunged in known weights of water and the rise of temperature of the water was noted.

The trials were made on two boilers that had between them 1,076 square feet of heating surface. The ratio of the heating to the grate surface was varied from 35 to 30, and afterwards to 25.

The dimensions of the boiler under trial were:

Grate surface, square feet (at ratio of 25)	43.06
Heating surface of the tubes, square feet	1,041
Total heating surface, square feet	1,076
Ratio of heating surface to grate surface	25, 30 and 35
Section of side ash-pit doors, square feet	9.04
above brick wall, square feet	9 47
of passage for gases entering tubes, square feet	18
of passage for gases leaving tubes, square feet	9
of smoke pipe, square feet	10.22

Natural draft was used in all trials except the fifth and the sixth.

RESULTS OF TRIALS.

	First.	*Second.*	*Third.*	*Fourth.*	*Fifth.*	*Sixth.*
Ratio of H.S. to G. S.....	30	35	30	25	30	30
Temperatures *a*.....	863	674	726	831	903	924
b.....	389	554	567	632	506	627
e.....	382	581	.....	545	651	615
e'	693	.....	.....	.....	768	770
d.....	1660	990	1122	1585	1083	1145
m.....	370	.....	466	512	427	510
Length of trial, hours.....	6	6·45	6	6	3	3
Total coal burned, lbs.....	2216	3146	3326	3960	2693	3326
Weight of residue, lbs.....	176	267	512	336	229	285
Pressure of steam, lbs.....	50	43	50	50	48	43
Temperature of feed.....	77	77	70.6	80.5	69	70
Coal per sq. ft. grate.....	10.31	15.1	15.46	15.3	25	31
Steam per lb., coal.....	10.67	9.58	9.23	8.97	8.02	8.75
combustible	11.65	10.47	10.92	9.8	8.763	9.652
Temperature of air.....	71	79	79	82	77	77
Draft in inches of water:						
At base of stack.....	.....	.....	.....	0.16	.....	.....
Combustion chamber.....	.....	.....	.....	0.10	.....	.....
Furnace.....	.....	.....	.....	0.08	.....	.....
Ash-pit.....	.....	.....	.....	0.02	.....	.....

In the third trial considerable air must have entered the boiler from the other boiler connecting with the combustion chamber, as the central doors to combustion chamber were closed.

RESULTS OF TRIALS (OLD BOILER).

For these trials the same boiler was used with the combustion chamber removed. The sides and the ends of the boiler were bricked up, and bafflers were interposed among the tubes to prevent the gases from rising directly to the stack.

The trials were under the auspices of French naval engineers, as were those preceding.

Natural draft was used in all the trials of the old boiler.

	First.	*Second.*	*Third.*
Ratio of H.S. to G.S.....	30.	30.	30.
Temperatures, *a*	460.	509.	510.
b	1,006.	965.	670.
e	1,274.	1,119.	1,005.
e'	590.	748.	656.
m	590.	568.	670.

	First.	*Second.*	*Third.*
Length of trial, hours	6.	3.	3.
Pressure steam, pounds	50.	52.5	53.5
Temperature of feed	73.	72.4	74.
Coal per square foot of grate	15.55	15.58	20.6
Steam per pound of coal	6.725	5.436	6.526
Temperature of air	77.	77.	79.

In all cases there was a thick black smoke at the time of firing and for some time after that. The flame in the furnace was reddish, showing incomplete combustion.

COMMENTS ON THE PRECEDING TRIALS.

It was noticed that the combustion of the gases was much better in the first trials. The amount of carbonic oxide was much less.

It was also noted that the gases lost much more of their heat in the first trials than in the second set of trials. The temperature at the base of the stack (*m*) was less in the first trials. The economical results were so much superior in the first set of trials that no comment seems necessary.

It would seem that the presence of a combustion chamber on a tubulous boiler is a good thing.

From the results of the first set of trials it appears that 1 pound steam can be produced

At 10.2 pounds per square foot grate with a weight of 7 pounds.
" 15.3 " " " " " " " 4.94 "
" 25 " " " " " " " 3.81 "
" 31 " " " " " " " 2.86 "

The evaporation per square foot of heating surface is:

At 10.2 pounds per square foot grate per hour, 39.6 pounds.
" 15.3 " " " " " " " 55 "
" 25 " " " " " " " 72.6 "
" 31 " " " " " " " 96.8 "

CARE AND PRESERVATION OF THE D'ALLEST BOILERS ON THE *LIBAN*.

The first sea-going vessel to be fitted with D'Allest boilers was the *Liban* of the Fraissenet line. This company is the one that

owns the boilers, Mr. D'Allest being the engineer for the company.

The D'Allest boilers were placed on the *Liban* in 1890. At that time the engines were changed from compound to triple-expansion by adding a high pressure cylinder forward of the old engine. The *Liban* has performed the service to the Gold Coast and to Constantinople from Marseilles. Lately she has been put on the Corsican service. I made a trip in her to see the running of the boilers. The results following are what I saw then. The instructions of the manufacturer for the care of the boilers follow:

It was not possible to obtain any figures of the power of the ship that were at all accurate, and, in consequence, there was no possibility of obtaining any results of the boilers' performance. The only results to be obtained were those of the manner of managing the fires, and the general care of the boilers.

Raising Steam, Getting Under Way.—If the brick work is newly made it is dried before the fires are lighted for getting under way. This is done by keeping a light fire under the boiler for about four hours.

When the brick work is dry, the grates are covered with a layer of coal of as nearly a uniform thickness as possible. Fires are lighted as in the Scotch boilers, and are not forced. The heating of the boiler must be slow. The method of construction makes the danger from unequal expansion of the different tubes very great. In general, about four hours is taken to raise steam, but it is better to allow six hours for this purpose. The fires can be so pushed as to raise steam in one hour, but there will be many leaks at the plugs in the tube sheets, and at the tubes. These may take up after the boiler has been under way some time, but there is danger of their becoming so great as to necessitate the drawing of the fires under the boiler.

The feed pumps are started only when the steam is raised. The quantity of steam taken to the engines is regulated solely by the throttle valve, and the stop valves are opened wide on all the boilers. This is to ensure the same pressure in all the boilers.

Generally, when the fires are lighted there will be several leaks at the plugs in the tube sheets, due to the fact that the rings used

to make the joint between the plugs and the sheet are not usually good conductors. As soon as the temperature of the boiler is practically the same in all parts these leaks will take up; or they may be stopped at once by setting up on the screws that secure them in place.

Working the Fires.—In working the fires, great use is made of the dampers at the base of the uptake to regulate the draft. As the steam rises the dampers are closed, and *vice versa*. When the fires are fresh the dampers are wide open, and as they are burned the dampers are closed.

Care is taken with the firing. The layer of coal on the grates must be equal in thickness in all parts to secure the maximum efficiency. This is not so necessary in this boiler as in the Belleville, on account of the bringing together of all the gases before they pass to the uptake. But it is always an advantage to keep the fires in their best condition. Great care is taken to keep the stokers from firing only on the front of the grates, and the thickness of the fires is kept as low as possible without leaving holes in them.

The furnaces are charged regularly at intervals of ten minutes, and the furnaces that are coupled together are charged five minutes apart. This rule is not kept by a watch, but it is observed as nearly as may be. The two furnaces that are coupled together (by having a common combustion chamber) are never fired at the same time. The doors to the combustion chamber are kept closed, as it is found that enough air reaches the fires by the ash-pit doors to give good combustion of the coal. Every quarter of an hour or so the combustion chamber is examined by one of the peep holes on the front of the boiler, and care is taken to have it full of flame. Too much air makes the combustion chamber fill and empty of flame. Too little air gives a reddish flame that is easy to recognize.

Feed.—In this boiler, as in all others of the water tube type, the water must be fresh. Mr. D'Allest makes no claim that his boiler can be worked with salt water, differing from Mr. Belleville, who says that the use of salt water is not dangerous, but requires some special care to be taken. Mr. D'Allest says that

the use of salt water is absolutely dangerous, and that the presence of oil or greasy matters in the feed water is equally dangerous. This is due to the amount of tube corrosion that occurs with the use of salt water, and from the presence of oils in the boiler. The boiler should be filled in the first place with distilled water, but the use of fresh river water is allowable. For making up waste feed the use of distilled water is necessary. A complete distilling plant is fitted on the *Liban* for this purpose, steam for the distiller being taken from the first receiver, and the steam produced sent to the second receiver.

No oils except mineral oils are used for the piston rod, as well as for the cylinders and the valves. A filter is fitted on the pipe between the air pump and the feed pump, and lime is added to the feed water in the same way as on the *Australien*. Mr. D'Allest gives as the reason for the use of lime in the feed water the quantity of salt water that must inevitably enter the boiler from the condenser. He also says that the lime combines with the oils and grease in the feed water, and that this is precipitated for the most part at the bottom of the rear water leg. The amount of lime used on the *Liban* is about six pounds per day per 1,000 I.H.P.

The lime is introduced in a small tank placed abaft the filter, and connected at the top with the discharge of the feed pumps, and at the bottom with the main feed tank. A fine sieve is placed near the bottom of the tank to keep the lime from entering the feed water in a solid form.

The feed pumps used are of the Thirion type, which are used on all ships in the French Navy not fitted with Belleville boilers. This rule is applicable to the French merchant marine also. Of course, there are exceptions, but this is the standard pump in France.

Use of the Blows.—Every eight hours both the surface and the bottom blows are used. The saturation is limited to 3 or 4, and preferably to 2.

General Working of the Boilers.—In this respect there is no difference to be remarked between these boilers and Scotch boilers, except that the smaller quantity of water in them makes the

liability to variations of the steam pressure greater. It is also slightly more difficult to keep the water level. The ordinary check valve is used to regulate the feed water, and the water level has to be watched carefully, although there is no trouble in keeping it constant. Great care must, however, be taken with the firing. This must be regular. It is by care with the fires that the steam pressure is kept constant. If the firing is not regular the pressure may rise beyond the gauge pressure, or it may fall very low.

Salt Water Leaks in the Condenser.—The only legitimate cause of salt water in the boilers is a serious leak in the condenser. This must always be kept clean. If there is a leak in the condenser that cannot be repaired before using the boilers, the following precautions must be observed:

The dose of lime must be tripled.

Only a part of the boilers will be used, and the fires will be forced under them to give rapid circulation in the tubes, and so to prevent the deposit of salt in them.

The blows must be used a great deal.

Cleaning Tubes.—The tubes being placed behind a water leg there is great difficulty in sweeping them from the front of the boiler. The hollow stays in the front water leg may be used for a steam jet; this has been tried, but has not given good results.

A plan for sweeping the tubes from the side has been tried on the *Liban* with success, and will be fitted to all future installations of D'Allest boilers for the French service. This consists essentially of a steam tube at the middle of the side of the boiler farthest from the combustion chamber; it has universal joints at both ends, and can be turned through a half circle so as to turn the small tubes fitted on the central tube between each row of tubes through the tubes. Thus the small tubes are so placed as to cover most of the tube surface. By turning on the steam to the central tube, each of the small tubes acts as a steam jet, and as it is turned around it serves to sweep the tubes. Holes are fitted in the small tubes pointing in all directions, up, down and ahead. When the central tube has been turned back and forth five or six times the boiler is practically clean. The soot goes up the

chimney for the most part, but some of it settles at the bottom of the uptake and is shovelled from there. This operation takes about three minutes for each boiler, and takes about four inches of water from the boiler. The tubes are cleaned every two days.

The tubes may also be cleaned with brushes of a special form. This brush consists of wires seized between two steel sheets and arranged in a layer about $\frac{3}{4}$-inch thick. It is introduced between two rows of tubes and turned through a right angle. Then it is pushed the length of the tubes. This can be done when the boiler is in use, but only with great difficulty. The sweeping of the tubes with the brush has not given as great satisfaction as the use of the latest scheme for sweeping tubes by steam.

When the ship comes into port the tubes are always swept by steam to use up spare steam.

Coming to Anchor.—The quantity of coal on the grates is made as small as possible when coming to anchor. All doors are closed, and the boilers are left to cool off as with ordinary boilers. The spare steam is used in sweeping the tubes and in running the winches. The blows are opened wide several times to reduce the concentration as much as possible. In this way it will rarely be found necessary to blow off steam to the condenser.

Cleaning Inside of the Boiler.—In spite of the use of the filter, there will always be found more or less grease in the boilers and the tubes may be entirely covered with a scale of grease and lime and salt. To guard against this the boiler is washed every three months with a solution of caustic soda. This washing is very easily done, and often avoids scraping the inside of the boiler. About 11 pounds of caustic soda are introduced in the boilers per ton of water contained. The soda is introduced in a liquid form and by the feed pumps. The concentration of the water in the pumps is not allowed to exceed 1 pound of caustic soda for 7 quarts of water, in order to avoid danger of attacking the brass of the pumps.

When the soda is in the boiler a light fire is started and kept up for two or three hours, a pressure of 30 to 40 pounds being maintained. Very little steam must be made and the fires are consequently very light. The surface and bottom blows are then

used and the boilers are left to cool down. When cold, they are completely emptied by the cocks on the bottom of the water legs. It will be found that there is a large amount of oil sent off with the water.

While the boiler is being emptied the man-hole plate on the steam drum is taken off and the drum cleaned by simply sweeping it with a broom. It is important that the cleaning be done as soon as the water has left the drum so that any sediment in the drum can be removed while soft.

Some of the tubes are then opened, especially those over the combustion chamber. It is not often that there is any necessity to clean these tubes, but if at all dirty they are cleaned by passing a metallic brush through them. One or two tubes of each row are then opened for inspection, and finally all the tubes of the lowest row are opened and carefully cleaned. The bottom of the water legs is cleaned also.

If there is any salt in any of the tubes it must be removed. The tubes are scraped by means of a special scraper. The water legs are scraped, if any salt is found on them, from the holes opposite the tubes in the front sheet of the boiler.

Repair of the Brick Work.—The bricks used in the boiler are not cemented together, but simply placed side by side. Those of the arch above the combustion chamber, and those above the upper row of tubes will last almost indefinitely if only left alone. Some of those in the lowest row may be broken when cleaning the boiler, but they can easily be replaced. These bricks are ordinary fire bricks, cost little, and can easily be replaced when broken.

The sides of the furnaces and the brick walls are of ordinary fire brick laid against each other and covered with fire clay.

Preservation of the Tubes.—The only tubes that require any special attention are those of the lowest row, immediately above the fires. If they are kept absolutely clean they will not burn out and will not bend. But it is not possible to keep them quite clean. There is always more or less greasy deposits in them. They will then bend with the concavity turned towards the fires, and will leak at the joints in the tube sheets. These leaks are

not of any importance, and can be easily stopped by expanding the tubes. Tubes are also sometimes burnt. This occurs only with the tubes of the bottom row, or to those tubes above the combustion chamber. The breaking of a tube may be foreseen by the presence of small reddish spots on the inside of the tube. If any of these spots are seen when cleaning the tubes, the tubes must be changed. These spots mark places where there will soon be a pocket, and where the tube will be burned. A broken tube has, however, been known to run for forty hours without the size of the break appreciably increasing.

The tubes that best resist the action of the fires are the Serve tubes. They are stronger to resist bending, and consequently less liable to give trouble from leaks; also, even if they are covered with a deposit, they may not burn, as the projections are large enough to conduct the heat from the tube to the water. For these reasons Serve tubes are used exclusively for the lowest row.

If the water level gets too low the tubes at the top of the combustion chamber will become empty and therefore more liable to burn. Such an accident happened to the boilers of the *Liban* when they were first tried. Two men were killed. Since that time there have been three rows of tubes over the combustion chamber, and these have more recently been Serve tubes.

Where the power of the boiler is too great for that to be expected from the heating surface of the boilers, the more Serve tubes there are the better, as the efficiency of the heating surface is increased by their use. In projected designs of these boilers, Serve tubes will be used for all tubes in contact with the flame.

Changing a Tube.—If it is a tube of one of the outside rows of tubes that requires changing, it can be reached by the furnace or by the sides of the boiler, and may be cut near the tube sheet. The tube is then drawn out, and also the small ring that remains in the tube sheet, but this last after the rings are deformed by a hammer.

If, however, the tube to be removed is in the middle of the nest of tubes, it cannot be reached except by the holes in the ends of the boiler opposite the adjacent tubes. The ends of the tube are

deformed as usual. The chisel used is of a special form calculated not to injure the boiler sheets. Generally it is sufficient to deform only one end of the tube and to force the tube from the back end by a tube driver.

A new tube is put in place and expanded in the ordinary way.

Plugging a Tube.—No tube should be plugged by a stay, as the stay not being in the water will expand when heated and the plugs will no longer be of any use.

It may also be stated that it never pays to plug a tube if there are any spare tubes on the ship. The time required to plug a tube is as great as that required to change the broken one and replace it by a new one. This is aside from the advantage of having a full set of tubes in the boiler.

A tube may be plugged by screwing on the face of the tube a plug of the same form as that for the outside sheet of the boiler. The screw that holds the plug in place is held by the plug in this outside sheet.

Leaks at the Plugs in Boiler Front.—There is frequently a case of a leak at one or more of the plugs in the outside plates of the water legs. These are caused by the introduction of a quantity of cold water in the boiler, or by the too rapid cooling or heating of the boiler. The rings of asbestos, used in a large measure to make these joints, are not good conductors of heat, and the temperature of the plugs is not the same as that of the boiler when that of the boiler is changing. The use of the improved rings of copper and lead has done away with most of the leaking at the plugs. If there is any leak at these joints, all that is necessary to do is to wait till the boiler has attained a fixed temperature, after which the leak will stop.

Boilers not in Use.—When a boiler is not to be used for some time it is filled completely, after being washed, as already described. The water in the boiler is fresh water, with about 22 pounds of caustic soda added per ton of water. This is to avoid all internal corrosion. All doors are tightly closed to avoid drafts in the boiler, and so to avoid any external corrosion.

For this reason it is best to cover the smokepipe.

General Instructions.—The boilers must receive only fresh water

to fill to working level, and to make up waste feed. Consequently, loss of fresh water from the boiler must be guarded against, and the condenser must be in good condition. A distiller must be used, and lime be added to the feed to destroy the effects of the oils and salts in the water. This will also reduce "priming." A filter to remove the grease from the feed water is a necessity. Only the best mineral oil can be used in the cylinders.

In starting the fires, they should not be forced at all. They should not be pushed until the engines are well under way. While under way, the grates should be covered regularly and with as thin a layer as possible. Care must be taken not to charge the front of the grate to the exclusion of the rest.

When stopping firing, there should not be a large amount of coal on the grate, and the doors should be closed, and the boiler cooled down as slowly as possible.

The two furnaces coupled together on the same combustion chamber must never be fired at the same time.

The tubes of the lowest row must always be kept absolutely clean. Every three months or so, if the boiler is in constant use, the washing described must take place.

The brick work must be kept in good condition and all broken or lost parts replaced as soon as possible. The arch above the combustion chamber must be kept in good condition.

With proper care in managing the firing, the safety valves will never open. All that is required is proper care when getting under way and coming to anchor.

Specialties in the Design of the Boiler.—No solid drawn tubes have been used.

The boilers have zinc plates in all accessible parts. The quantity is fixed at about 2 pounds per 1,000 square feet of heating surface. The use of zinc plates is found to be a very good thing in these particular boilers. Mr. D'Allest said that it was absolutely essential, and that with it the boiler would last double the time it would without it. The plates are put in a box of the same form as that used in our Navy. Two plates are put at the bottom of each water leg, and two are placed in the steam drum. Those in the water legs are placed on the outside sheets, so as

not to interfere with the circulation of the water in the tubes. The plates in the rear leg are not placed at the bottom, but opposite the second row of tubes from the bottom.

In placing these boilers in a ship, space is always left at the sides for getting at the tube nest. This space does not exceed that necessary for the uptakes.

OFFICIAL TRIALS OF THE D'ALLEST BOILERS OF THE *BOUVINES*.

Natural draft was used in the first two trials of these boilers. In the last two trials the draft was assisted by blowers. The third and fourth trials were in succession. The grates were not cleaned between these trials, which were the official trials for the acceptance of the boilers.

The ratio between the heating and grate surface was 30.

	First trial.	*Second trial.*	*Third trial.*	*Fourth trial.*
Length of trial, hours	6.	12.	3.	3.
Coal burned per square foot of grate, pounds	20.5	12.3	24.5	30.7
Temperature of feed	119.5	123.5	125.	127.
Pressure of steam, pounds	185.	185.	185.6	185.6
Evaporation per pound coal, pounds	9 8	10.068	9.501	9.235

TRIALS OF THE *JEMMAPES*.

The engines of the *Jemmapes* were designed for 8,400 I.H.P. with forced draft, but the official trials gave 9,250, and this with a consumption of 2.03 pounds per I.H.P. The coal burned per square foot of grate was 29.66 pounds.

On another of the trials the cost of an I.H.P. was (7,711 I.H.P.) 1.802 pounds. The coal per square foot of grate in this trial was 22.05 pounds per hour. The second trial was of twelve, and the first of six hours' duration. The results of the other trials were not public when I left France.

All D'Allest boilers are designed by the inventor, though most of them are manufactured by other firms than the one of which he is engineer.

THE NICLAUSSE BOILERS.

These boilers are on the principle of the Field tubes. Each of the evaporating tubes is fixed at one end in a water leg with

two divisions, one for steam and one for water. To carry the water from the water side of the leg there is a tube fitted connecting the water leg to the end of the evaporating tube. The water then turns in the tube and passes through the outside or evaporating tube to the steam side of the water leg.

The features of the boiler that are peculiar are the way of making the joints between the tubes and the water leg; the form of the tubes and the water leg, and the kind of nappe on the top of the steam side of the water leg at the point where the steam enters the drum.

The tubes are all of welded steel, and are reinforced at the joints which are always conical, whether the joints are screw or not.

The joint between the evaporating tube and the water leg is a flat conical one at both the outside sheets of the water leg. The joint at the partition between the steam and the water sides of the leg is a loose one, not water tight. The cones forming the joints to the outside sheets are shrunk on the tubes and are then turned to the required diameters. The cones are not exactly the same, as in placing the tube in place the cone on the inside sheet of the water leg begins to bear last. The cone on the tube for the outside sheet is made comparatively flexible by being secured at one end only. The inside diameter of the outside cone is made the same as the outside diameter of the inside cone. The tube can be drawn out, and it is in equilibrium for the steam pressure. The tube does not tend to leave the water leg. Nevertheless, a safety bar is screwed on the outside of the water leg to ensure the tube staying in place in case of shocks, etc.

The end of the evaporating tube is in the form of a lantern with a section at the middle partition of the water leg that is approximately the same as that of the tube. The diameter of the hole in this partition is very slightly larger than that of the tube at the inner cone.

The inner tube which serves for the circulation of the water to the end of the evaporating tube is screwed into the outside of the latter. The joint is conical. The tube has a small lantern

to form the junction between the two parts of the tube. The tubes are shown in the accompanying sketches.

The nappe at the top of the steam side of the water leg is made conical with the large diameter at the top of the nappe so as to give a small velocity to the steam when it is entering the drum at the top of the boiler.

The disposition of the nappe gives dry steam at rapid firing of the boiler.

The collectors are of malleable iron, and there is a separate collector for each pair of vertical rows of tubes. The tubes are arranged staggered. This is to give the gases of combustion a zig-zag course through the tubes. Space is left between the collectors to sweep the tubes by steam. The joints between the collectors and the drum are conical, and the drum is fastened to the collectors by bolts, to make the joints tight. The collectors are united at their lower part to enable the blowing down under pressure. There are pipes to make this connection.

The feed water is introduced in the upper drum in a stream going in the steam. No ejector is fitted to the boiler, and no part of the boiler is particularly apt to have more sediment than another unless it be the bottom of the collectors where the bottom blows are placed.

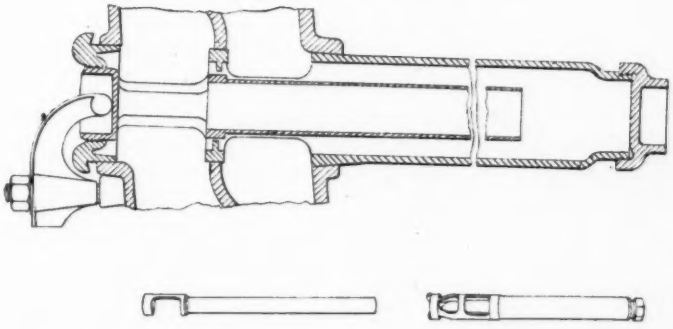
It seems to me that there is too much machine work on the tubes. This makes the boiler costly. The conical joints are satisfactory and appear to be good. I think, however, that the joint between the collector and the drum might as well be of the ordinary form.

The circulation of the feed water in the boiler depends entirely on the difference in weight of the different columns in the water leg. The higher the boiler the better the circulation. The circulation is, however, good.

The economy of the boiler is not great at high rates of combustion. I think if a combustion chamber were fitted the economy would be as great as that of the D'Allest boiler, if not superior.

The great advantage of this boiler is the freedom of expansion of the tubes and the consequent ease of forcing. When forced, the tubes bend, and I suppose that they will wear out sooner than if Serve tubes were used for the bottom row.

Section & Views of Evaporating
& Circulating Tubes
Niclausse Boiler



The inner tube is not supported and will sag so as to bear on the evaporating tube, but it is in a constant state of vibration and this would not necessarily be an objection of weight.

The floor space occupied for a given heating surface is here reduced to a minimum. The tubes are slightly inclined but the ends of the boiler are vertical.

All work on this boiler can be done from the front of the boiler, and no space is necessary at the sides or back.

The boiler does not work regularly enough, due to the small amount of water in it. But it can be worked without any auxiliaries such as are needed on the Belleville boiler, and will give fair satisfaction even when used alone without Scotch boilers.

It is very easy to repair, and the repairs are quickly made.

The inventor claims the advantages of small weight and good utilization of coal in addition to those given, but the weight is slightly greater than that of the D'Allest boiler, and is much greater than that of almost any type of bent tube boilers in use in this country. The weight may be reduced, but it is already a great deal less than that of the Scotch boilers whose place it is trying to take. The advantage of using straight tubes lying in the ease of cleaning and repair, overcomes the advantage of the smaller weight of the bent tube boilers.

The inventor claims that the impossibility of emptying the tubes except when cleaning the boiler is a great advantage. He says that this retains whatever deposits may be made in the tubes in a soft muddy state that facilitates the cleaning wonderfully.

The tubes and all other parts of the boiler are made interchangeable. The boiler is in small elements and can be taken out of a ship with ease. Any part that requires replacing can be changed without great cost, owing to its small size.

Care of the Boilers.—The inspections of the tubes will be more or less frequent according to the quality of the water used, and the amount of work done by the boiler. When the boilers are first used the tubes should be examined frequently, especially those of the lowest row. This is to determine when and how often the boiler should be cleaned.

The part of the collectors that is most liable to become dirty

with use is the upper part. Cases have occurred when the deposits in the top of the collectors have stopped the circulation, and tubes have been burned out.

The boiler should not be emptied except when about to clean it, as the deposits would otherwise become hard and much more difficult to remove. The deposits will be found to be in most cases in the circulating tubes to the exclusion of the evaporating tubes.

Both the surface and the bottom blows should be used at least once a day. The cocks are very liable to get choked, and must be frequently examined to see that this does not occur.

To clean a tube, it is necessary to remove the safety bar, and to unscrew the inner tube. These tubes contain the deposits, which are removed by washing, or, if it is necessary, by using a metallic sponge. A special key is furnished to prevent the turning of the outside tube when the inner one is unscrewed. This turning of the outside tube is not frequent. If the evaporating tube needs cleaning, it can be done either in the boiler or it can be drawn, the screw plug at the back taken off, and the tube cleaned outside the boiler. To draw the evaporating tube, it is started with a hand hammer. Before replacing a tube in the boiler, the joints are carefully cleaned and covered with grease.

In general, all the tubes of a boiler are cleaned at the same time, commencing with the bottom row in removing them, and with those of the top row in replacing them. While the tubes are out of the boiler the collectors are wiped and cleaned.

The tubes are cleaned on the outside by a steam sweep that is placed in the openings between the collectors. They are swept in horizontal rows, beginning at the top of the boiler.

Other parts of the boiler requiring to be examined from time to time are the feed pipe and the fittings of the drums.

Boxes to collect the deposits from the feed water are put in the drum, and must be cleaned frequently.

The feed water is regulated entirely by a check valve placed on the boiler. Any sudden increase in pressure is stopped by use of more feed water.

In case of leaks, it is not well to screw up on the nuts on the safety bars. This may break the joint and cause an explosion of the boiler. All leaks are the result of dirt in the joint, and no leak can be stopped without remaking the joint. If the joints are properly made in the first place, no leaks will appear during the working of the boiler.

During stops, the ash-pit doors are closed, but neither the furnace doors nor those to the tubes are opened. More especially if the feed water falls below the bottom of the glass should these doors be closed. In this case, the fires must be drawn at once unless the cause of the lack of water is at once discovered.

When the boilers are no longer required, the blows are used and the water level is raised to the top of the glass. The fires must be pushed to the back of the grate or even drawn. Sudden changes of temperature will have no bad effects.

When a boiler is not to be used for some time it is cleaned and then completely filled with water.

EXTRACT FROM A CIRCULAR FROM THE MINISTER OF MARINE CONCERNING THE USE OF LIME IN THE FEED WATER.

According to the opinion of the Inspector General of Engineers it is best to add lime to the feed water after it leaves the filter, as, for example, in the feed tank, but care must be taken to prevent the lime from settling at the bottom of the tanks. It is best, on the contrary, that the lime should go to the boiler and mix with the water there, and be dissolved. In certain cases it may be advantageous to introduce the lime directly into the boilers. In this way it would not pass through the feed pumps. As to the amount of lime to use, it is better not to use too much, and for this reason, it is essential to determine at least once a watch the acid or basic condition of the water in the boilers by using litmus paper, or in some other simple way.

Until further orders, the amount of lime to be delivered will be 1.7 pounds per ton of coal in the bunkers. But it is understood that the amount to be used will vary so that the water in the boilers will never have any trace of acidity, but give a good basic reaction.

COMPARISON OF THE ENGINES AND BOILERS OF THE *BUGEAUD*, *CHASSE-LOUP-LAUBAT*, AND *FRIANT* FITTED WITH BELLEVILLE, D'ALLEST AND NICLAUSSE BOILERS.

These ships are exactly the same except in the fire-rooms. They were intended to give a comparison of the three types of tubulous boilers used on them.

The engines of these ships all come from the shops of the Forges et Chantiers de la Mediterranée, at Havre, and are the same in all particulars.

The boilers come from the inventors of them.

The hulls of the *Bugeaud* and *Chasseloup-Laubat* come from the navy yards at Cherbourg; that of the *Friant* from the navy yard at Brest. They are identically the same.

The ships are second class cruisers of 3,725 tons displacement. Length, 308 feet; beam, 43 feet 6 inches; depth below main deck, 29 feet 6 inches; draught, 17 feet 6 inches at midship section, and 20 feet 8 inches aft; area of the immersed midship section, 697 square feet. The lines of these ships are derived from those of *Davout* that gave satisfaction. The ships are designed to give a speed of 19 knots with forced draft of 1 inch water pressure in the fire rooms. They have each a crew of 356 men, all told, and were all launched in 1893.

The designed power of the engines at forced draft was 9,000 I.H.P.

The contract calls for the following powers and dimensions of engines and boilers for the three ships:

The total power of the main engines at the forced draft trial is 8,650 I.H.P. If the number of revolutions is less than 135 it will be increased to that figure. If at 135 revolutions the power (with a screw given by the Government) is less than 9,000 I.H.P. the dimensions of the screw may be changed to increase the power.

The price of the engines of each ship is 1,314,700 francs. The price of the boilers is for the Belleville boilers of the *Bugeaud*, 603,570 francs; for the D'Allest boilers of the *Chasseloup-Laubat*, 503,300 francs; for the Niclausse boilers of the *Friant*, 615,170 francs.

The total weight of all machinery is to be (boilers and water

included), *Bugeaud*, 801 tons; *Chasseloup-Laubat*, 757 tons; *Friant*, 760 tons. For any excess in weight within 5 tons a penalty of 1,000 francs per ton is charged; for excess within 10 tons, 2,000 francs per ton is charged; for excess within 15 tons, 5,000 francs per ton is charged; if in excess above 15 tons the machinery may be rejected.

There are to be the following trials:

One forced draft trial of four hours, with a combustion of not more than 30.7 pounds per square foot of grate. The power on this trial must be as much as 9,000 I.H.P. for all machinery. If on this trial the consumption of coal per square foot of grate is as great as 27.7 pounds, no second forced draft trial is necessary; if it is not, there will be a forced draft trial with a consumption of coal at least equal to 30.7 pounds per square foot of grate per hour.

There will be a trial of twenty-four hours, of which four hours will be at about 6,000 I.H.P., two hours at 9,000 I.H.P., and eighteen hours at 6,000 I.H.P. This last period may be taken at any part of the trial. The grates will be clean at the commencement of the trial, and the coal burned per I.H.P. will be from 1.76 to 1.98 pounds per hour.

There will also be three trials of six hours each, with powers of 7,000, 3,500 and 1,500 I.H.P., and with coal consumptions of 1.87 to 2.09 pounds; 1.54 to 1.76 pounds, and 1.65 to 1.87 pounds respectively. If the consumption exceeds 2.21 pounds on the first trial, or 1.98 pounds on either of the others, the machinery may be rejected.

In each of the trials where the consumption is fixed, any excess involves a fine of 100 francs for each hundredth pound. There is an equal premium for all saving above the figures given. No premium is given for any excess of power, and no penalty is imposed for a failure to arrive at the required power. It is understood that the machinery will be rejected if it fails to reach the required power.

DETAILED WEIGHTS IN THE ENGINE ROOM.

	Tons.
Engines complete, (with reducing valves)	246.9
Line shafting	51.0

	Tons.
Propellers	12.0
Bilge pumps	1 6
Piping and water valves	49.0
Floor plates and ladders	8.6
Tools	5 2
Spare parts	8.0
Evaporators	3.0
Engine room ventilators	2.4
Water in condensers and pipes	10.0
Water in tanks	5.0
Lime tanks	1.3
Total weights in engine room	425.0

For the *Bugeaud* (Belleville) the weight of water in tanks is increased to 16.0 tons, making the total weight for this ship 436.0 tons.

DETAILED WEIGHTS OF BOILERS.

	D'Allest.	*Belleville.*	*Niclausse.*
Boilers proper	131.3		146.5
Uptakes	67.0	273.3	56.0
Accessories	11.8		6.6
Grates and fittings	19.1		18.3
Tools and spare parts	5.6	8.1	7.6
Feed pumps	5.2	11.4	5.2
Tanks	4.0	8.0	4 0
Smok epipes	17.0	23.5	19.8
Floor plates and ladders	9.0	9.0	9.0
Fire room ventilators	9.0	9.0	9.0
Air compressor		2.5	
Separator		5.2	
Water in boilers	53.0	16.0	53 0
Total of fire rooms	332.0	365.0	335.0
Total of all machinery	757.0	801.0	760.0

The difference in the weights of smoke pipes is due to the fact that the different contractors did not include the same height of pipe in their estimates, and the weights given do not include the same portion of the pipes.

The table is taken from the contract weights, and is not the same as the weights actually in the ships. The weight of the boilers of the *Friant* are given by the contractors as follows:

	Tons.
Boilers alone	202.61
Uptakes	10.76
Water in boilers	46.18
Tools and spare parts	7.26
Joint to smoke pipes	5.15
Total of parts mentioned	271.96
Adding weights of parts not included in above, as in contract weights:	
Feed pumps	5.2
Tanks	4.5
Smoke pipes	19.8
Floor plates and ladders	9.0
Ventilators	9.0
We would have a total weight in actual fire room	329.0

The increase in weight of the boiler proper, and the decrease in weight of uptake, together with the decrease in the weight of water in boiler are to be noted. The total weight remains about the same as before.

DIMENSIONS OF THE BOILERS.

	D'Allest.	*Belleville.*	*Niclausse.*
Number of fire-rooms	3	3	3
boilers	20	24	20
furnaces per boiler	1	1	1
Length of grate, feet and inches	6-8	4-7	6-8
Width of grate, feet and inches	5-4½	7	6
Total grate surface, square feet	732	755-2	782-8
Total heating surface, square feet	19,451	21,594	23,338
Ratio H.S. to G.S	26.6	28.6	29.8
Outside diameter of tubes, inches	3.25	3.23	3.23
Inside diameter of tubes, inches	2.91	2.86 and 2.60	2.97
Length of tubes, feet and inches	7-9	6-4	5-8½
Diameter of circulating tubes, inches	..	..	1 9/16
Number of tubes in vertical row	10	9	9
Weight of water, tons	53	16.2	46.2
Volume of steam space, cubic feet	1,879	784	830
Boiler pressure, pounds	214	242	214
Pressure at engines, pounds	170	170	170

The excess in weight of the Belleville boilers is partly explained by the short grates on those boilers. The Niclausse boilers have a much larger heating surface than the D'Allest for the same power, and considerably more than the Belleville. This

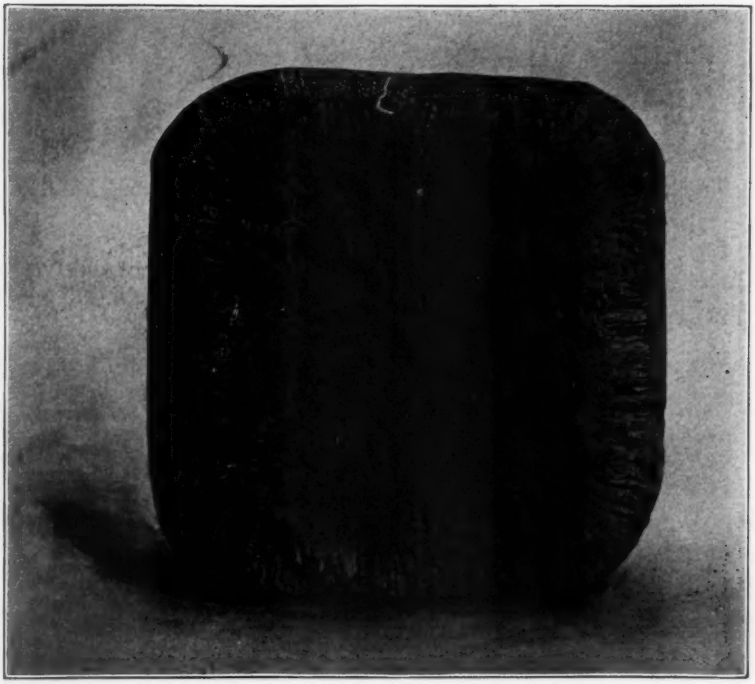
is while retaining a weight only equal to that required for the D'Allest boilers, and while having 40 tons less weight than the Belleville.

The ease with which the D'Allest boilers can be forced makes them far the most powerful boilers in an emergency of the three sets given.

There are two air pumps for each engine with diameters of 24.6 inches and stroke of 15.75 inches. The number of turns of the air pump was designed at 168 at full power. The air pump turns the circulating pump also. The diameter of the turbine is 47.2 inches.

The ratio of the final volume occupied by the steam to its volume on admission to H.P. cylinder is 7.4129.

Only fresh water is to be used for feed. A filter is placed between the engines and the boilers, and there is an apparatus for adding lime to the feed water. There are two feed pumps for each fire room, of which one is large enough to supply the boilers. There are two evaporators to supply waste feed. To make the working of the fires easier, there is a damper at the base of each smoke stack near each boiler.



FRACTURE OF OPEN-HEARTH STEEL INGOT.

AREA, 6½ BY 6½ INCHES.

Number of cells immediately under surface exposed to fracture, 253. Average depth of cell, 1⅛ inches. Average diameter of cell, ⅛ inch.

NOTES.

PHYSICAL REASONS FOR RAPID CORROSION OF STEEL BOILER TUBES.

The following statement by Mr. W. H. Gibbons, president of the Parkersburg Iron Company, on the above subject is from the "American Engineer and Railroad Journal":

In investigating the oft repeated assertion that "boiler tubes made from steel corrode, and become unserviceable much more rapidly than those made from charcoal iron," we have made the following experiments, as given below:

Taking a "heat" of ingots made of 7 inch × 7 inch × 4 inch, weighing about 650 pounds each, of the best open-hearth basic steel of following analysis: Carbon, .10; phosphorus, .014; manganese, .21; sulphur, .026; copper, .05, we first gave them a "wash heat," and cut them into two nearly equal pieces, then transferred them to heating furnaces again, and after another slight heating rolled them down direct to No. 9 gauge and sheared them into skelp for 4-inch tubes, being careful to keep separate the skelp made from tops and bottoms of ingots.

We found that the "bottoms" of the ingots invariably worked smooth and clean into plates, and sheared with only a normal wastage, but that the "tops" were almost uncontrollable in rolling, working soft, spongy and with much irregularity, and the surface of the plates when finished had a muddy, dirty appearance, indicating an excessive amount of cinder. Allowing one of the "tops" of ingots to cool after the first wash heat, on close examination its whole surface was found to be closely covered with minute holes, so close together that in a diameter of 1 inch as many as 25 of these minute holes could be counted, into which in many instances a needle could be entered to the depth of from 1 inch to 1½ inches.

Our next step was to nick and break with a heavy drop the upper or "top" of one of these ingots before heating, with the

result that the fracture developed an almost entirely spongy or honeycombed structure extending from $\frac{1}{8}$ inch under the skin or surface to about $1\frac{1}{4}$ inch in depth uniformly around the four sides of the ingot, and in many instances these cells ramified and extended to the center. Actual count of these cells in face of fracture $6\frac{1}{4}$ inches by $6\frac{1}{4}$ inches was 253.

This spongy condition no doubt revealed the at first unaccounted for difference in working between "top" and "bottom," as the same ingot when broken cold, half way up, showed much less of the cellular structure, which no doubt almost entirely disappeared on a nearer approach to the bottom, owing to hydrostatic pressure of molten steel in the mould driving out the gases before solidification occurred.

Our next step was to have tubes made from "tops" and "bottoms," with a report as to the working of the two, which only went to confirm our earlier investigations that the "tops" welded freely, and even blistered sometimes, while the "bottoms" worked hard and stubborn and were difficult to weld. Taking sections of these tubes made from the respective portions of ingot, they were put in a lathe, burnished, and subjected to microscopical examination, which revealed these same "cells" or "honeycomb" structures much decreased in size, but still present in great numbers in the "top" section, while the "bottom" was comparatively smooth and solid.

Each one of these compressed cells has walls of solid metal encasing infinitesimal shot of slag, which, in the boiler tube in service, with the incessant expansion and contraction of greater or less heat, will finally open slightly, admitting a little dampness, which under the heated condition of action will undoubtedly set up very rapid corrosion and early disintegration of the whole tube.

Before we had the opportunity of making this thorough test we were at a loss to know why steel tubes made from finest obtainable open-hearth stock should show such short lives when compared with those made from *charcoal iron*, but this would seem to be ample confirmation, for the known fact that, as a rule, steel has been most disappointing in service when compared with the

possibly less pure chemically, but more homogeneous and durable charcoal iron.

To make a free welding steel for boiler tubes, this sponginess *must* exist, and the more pronounced it is, the more thoroughly it will weld and the more rapidly it will corrode. Conversely, the more solid and free it is from "honeycomb" in the ingot, the more difficult, if not impossible, it is to weld, and the longer the *unwelded* life in the boiler.

It is a matter of record that the United States cruiser *Chicago* put new steel tubes into some of her boilers about two years ago, that were riddled with holes as large as shot inside of forty days' service, while others of her boilers had the original charcoal iron tubes that were put in when they were built, and which were still good after service of some five years.

COAL CONSUMPTION OF SOME BRITISH VESSELS.

During the discussion of a paper before the Institution of Naval Architects, Lord Charles Beresford stated that the coal consumption of the *Grafton* for a speed of 10 knots was 1½ tons an hour. From the following results obtained on a voyage to the Cape of Good Hope and back, and published in "Engineering," it appears that this figure is based on her consumption per I.H.P. for all purposes at 12.5 knots, and assuming 1,300 I.H.P. for 10 knots, which, of course, does not take into account the fact that the auxiliaries at the lower speed would tend to make the consumption per I.H.P. for all purposes greater at the lower speed. The following are the figures for the voyages:

	Outward.	*Homeward.*
Mean speed	12.46	12.54
I.H.P. for speed	2,500	2,500
Coal, in tons, per hour, making distance	2.46	2.44
including auxiliary engines	2.85	2.83
per day for all purposes	68.40	67.92
Pounds of coal per I.H.P., making distance	2.21	2.18
for all purposes	2.63	2.60

The *Grafton* is a first-class cruiser of 7,700 tons displacement. The *Crescent*, a sister ship to the *Grafton*, has completed two

voyages to Australia and back, the figures for which are given as 54,000 miles, 14,000 tons of coal, and a mean speed of 12.4 knots; which is equivalent to about 77 tons of coal a day.

It has been stated in the House of Commons that the *Sharpshooter*, since being fitted with Belleville boilers, has steamed for thirty consecutive hours, developing 2,326 I.H.P., on seventy-four tons of coal for the time.

Mr. J. I. Thornycroft has given, in the "London Times," the following record of the coal consumption trials of the *Speedy*:

Hours.	*Speed.*	*I.H.P.*	*Air pressure in tenths of an inch.*	*Coal per I.H.P.*
10	13	1,026	..	1.74
8	15	1,904	2	1.52
21	12.5	1,000	3 to 1	1.79
16	10	530	0	1.90
15	10	560	0	1.79
42	10.4	640	1	1.92
20	14.5	1,659	2.5	1.58
4	16.5	2,806	6	1.67
38*	7.3			

*Coal 812 pounds per hour.

SHIPS.

AUSTRIA.

Monarch.—"Mittheilungen aus dem Gebiete des Seewesens" gives the following particulars of this coast defense vessel, which is 305 feet long, 55.8 beam and 21 draught:

	Tons.
Hull, including 462 tons of deck armor	2,145
Armor	1,132
Machinery	836
Guns	600
Equipment	521
Crew and provisions	79
Coal at normal draught	197
Coal with full bunkers	492
Total weight with normal coal	5.510

She has two barbette turrets, each with two 24-cm. guns, six 15 cm. rapid fire in casemate, and eighteen smaller rapid fire guns.

The armor belt amidships is 10.6 inches thick, tapering to 4.7 at the bow, and terminating abaft the after magazines in 8-inch armor with an 8-inch bulkhead forward and aft at the end of the heavy armor. Above the belt is a citadel protected by 3-inch armor. There are two conning towers protected by 8-inch armor.

Her twin screw engines have been designed for 5,920 I.H.P. with natural and 8,380 with forced draft; they have cylinders $33\frac{1}{2}$, 51 and $78\frac{3}{4}$ inches diameter by $35\frac{1}{2}$ inches stroke, and work with a boiler pressure of 156 pounds. The air and bilge pumps are worked from the engines. The condensing surface is 11,840 square feet for the main, and 860 for the auxiliary condensers. Steam is furnished by three double and two single ended boilers, all 13.75 feet diameter; the double ones 18.63, and the single 8.83 feet long; the grate surface is 568, and the heating surface 15,770

square feet. There is one smoke pipe, the top of which is 63 feet above the grates.

The speed expected of this vessel is 17.25 knots.

BRAZIL.

The following dimensions of the five torpedo boats built by Schichau, and delivered to the Brazilian government last year, are from the "Zeitschrift des Vereines Deutscher Ingenieure":

Length on water line, feet	152.5
Breadth, feet	16.73
Draught aft, feet	8–2
Displacement, tons	136.
Coal in bunkers, tons	40.

The engines are of 2,200 I.H.P., and the speed on trial was as high as 28.3 knots, though the mean speed is not stated. They were designed for 26 knots.

ENGLAND.

New Cruisers.—The four first class cruisers provided for in the Navy estimates are to be similar in general design to the *Powerful* and *Terrible*, though somewhat smaller. Like the latter vessel, they will be sheathed with wood and coppered. Their dimensions will be:

Length between perpendiculars, feet	435
Length on water line, feet	455
Breadth, feet	69
Mean draught with keel, feet	25.25
Displacement, tons, about	11,000
Coal on above displacement, tons	1,000
Bunker capacity, tons	2,000

They will be named *Diadem*, *Europa*, *Niobe*, and *Andromeda*.

The armament will include fifteen 6-inch rapid fire guns, fourteen 12-pounders and twelve 3-pounders, besides a number of machine guns, and three torpedo tubes, two of which will be submerged.

The machinery will consist of two sets of triple expansion engines on each shaft, the combined I.H.P. of which will be about

20,000, which is expected to give them a speed of 20½ knots. The boilers will be of the Belleville type.

The four second class cruisers will be all steel vessels, and of the following dimensions:

Length on water line, feet	320
Breadth, feet	57
Draught, mean, feet	22
Displacement, tons	5,750

The battery will be the same as that of the vessels of the *Talbot* class. They will be named *Vindictive*, *Gladiator*, *Furious*, *Arrogant*.

The engines will be of 10,000 I.H.P., natural draft, giving a speed of 19 knots. The boilers will be of the Belleville type.

The two third class cruisers, *Pelorus* and *Proserpine*, will be "improved Barhams" of the following dimensions:

Length on water line, feet	300.
Breadth, feet	36.5
Draught, mean, feet	13.5
Displacement, tons	2,100.

The engines will be of 7,000 I.H.P., and the speed expected 20 knots. The boilers, as in the case of the first and second class cruisers, will be of the Belleville type.

The armament will comprise eight 4-inch and eight 3-pounders, besides torpedoes.

In addition to the above mentioned vessels, twenty torpedo boat destroyers, with a speed of from 30 knots, are to be built.

Bruiser.—This torpedo boat destroyer, a sister vessel to the *Ardent* and *Boxer*, described in the last number of the JOURNAL, was launched from the works of Messrs. John I. Thornycroft & Co., on the 27th February, and had a successful trial on the 29th of March, the following account of which is condensed from "Engineering":

The *Bruiser* steamed easily down to the Lower Hope, meeting the young flood, the wind being W.N.W., with heavy squalls and hail showers at intervals. The weather was so bad that is was decided to run the trial miles on the Lower Hope course, instead of going out to the Maplin, as is now usual. Seven runs were

made on the mile—the first, done in 2 minutes 6 seconds, being thrown out, the counters not being put in gear until the middle of this mile. The mean speed of the six runs was 28.14 knots. The rest of the three hours full speed running was made between the Mucking Light and Warden Point, in an awkward sea, the wind meeting the tide. The speed for the three hours was 27.97 knots. It is a curious fact that in this trial the boat only missed making the even 28 knots by rather less than her own length on each hour.

The average results for the measured mile runs were:

Steam pressure in boilers	204
Air pressure in fire rooms	2
Vacuum	26.2
Revolutions	376 6
I.H.P.	4,156
Speed in knots	28.14

For the entire three hours, the steam pressure averaged 202 pounds and the revolutions 374.3.

The coal consumed was 17¼ tons on the three hours' run; but it will be understood that it was not weighed as used—such a procedure would be manifestly impossible upon a full-speed trial of a vessel of this class—but it was carefully weighed before being put on board, and what remained over was estimated. It will be remembered that this was an official trial, so the records were taken by disinterested persons. The rate of combustion gave a fuel consumption of about 68 pounds of coal per square foot of grate per hour. In previous trials as much as 79 pounds have been burnt per hour per square foot of grate. The indicated horse-power on the six mile runs averaged 4,156, the trial being run with engines linked up.

At the conclusion of the speed trials, circles were turned on both hands. The circle with helm to port was accomplished in 1 minute 46 seconds, with the helm to starboard in 1 minute 52 seconds. This was in a strong wind, the force being 5 to 6. The weather was too rough to make circles astern, so that part of the trial is reserved for another time.

The machinery is the same as that of the other vessels of this

class constructed by the Messrs. Thornycroft, except that the boilers are like those of the *Ardent*, in which a slight increase in heating surface has been made over those in the *Daring*, and that the screw propellers are of manganese bronze, with blades cast with the hubs.

It has long been the practice of the torpedo boat builders to forge the propeller blades separately and key them into a boss. This plan answered well so long as steel blades only were used, but when manganese bronze was required, it was found there was danger of the blades getting loose in the boss. In order to avoid this, a new method of keying in was devised, but it is now thought desirable to have the propellers cast in one. The screws of the *Bruizer* are three-bladed, all surfaces being polished.

The furnace doors are fitted with springs so as to be self closing, and the ashpit doors being self closing, there is less danger to the stokers in case of a burst tube, as the steam would go up the chimney.

In the *Speedy* a new device was tried for controlling the distribution of the feed in the water-tube boilers with which that vessel was fitted. It is this question of feed distribution which was so long the rock ahead for the water-tubists when an attempt was made to run these boilers in groups. It was argued, not without reason, that in steam generators of this sensitive nature, and with so small a water-holding capacity, unless the feed were very evenly distributed, disaster would follow. In order to avoid this, various devices have been tried, with more or less success. That introduced by Messrs. Thornycroft consists of a hollow steel float, capable of withstanding the boiler pressure, which is placed inside the separator or steam-collecting cylinder which forms so important a feature in the Thornycroft boiler. By a system of levers, somewhat too complicated to describe without the aid of diagrams, the float regulates the opening of a check valve which is placed within the boiler, the amount the valve is opened determining the volume of feed. In this way the water level in the boiler determines the amount of feed admitted; thus, if the water level falls, the check valve is thrown wider open, if it rise the check valve is closed. Unlike most gears of the kind, the motion

of the float has not to be conveyed to the exterior of the boiler through a stuffing-box. An arrangement of the latter nature must lose much of its sensitiveness, and herein the Thornycroft gear has a manifest advantage. There is, however, in this device a means of regulating the normal water level by hand, and this is effected through a rod which passes outside the separator through a stuffing box, and is worked by a handwheel. In this way the gear may be set so that the water level can be carried at any required height. The gear has been found in practice to act admirably, and it is contemplated using it in some vessels built by other firms. On the occasion of the *Bruizer's* trial the feed water between the three boilers was properly distributed by it, the report being that no hand adjustment was required throughout.

The following is a resumé of the principal dimensions and trial data of the five vessels of this type built by Thornycroft and Company:

	Daring.	*Decoy.*	*Ardent.*	*Boxer.*	*Bruiser.*
Length on water line, feet	185.	185.	201.5	201.5	201.5
Breadth, feet.....	19.	19.	19.	19.	19.
Depth, feet.....	13.	13.	13.	13.	13.
Draught, mean, feet.....	6.28	6.25	6.67	6.67	6.67
Displacement, tons.....	240.	240.	280.	280.	280.
Revolutions per minute, three hours...	379.	364.	396.	410.5	374.3
Revolutions per minute, measured mile.....	389.	365.5	398.	412.5	376.
I.H.P., three hours.....	4,409.	4,049.	4,306.	4,543.	.....
measured mile.	4,644.	4,009.	4,343.	4,487.	4,156.
Speed, knots, three hours.....	27.706	27.763	27.973	29.175	27.970
measured mile..	28.213	27.641	27.840	29.080	28.144

The engines in all are of the twin-screw triple-expansion type, with cylinders 19, 27 and 27 (2) inches in diameter by 16 inches stroke. The boilers have 189 square feet of grate and 8,892 of heating surface in the *Daring* and *Decoy*, and slightly more heating surface in the others. The screws of the *Daring* are 6 feet 2 inches in diameter and 8 feet 9 inches pitch, and have a blade area of 13.5 square feet.

A coal consumption trial of the *Ardent* was made on the 26th of March, lasting twelve hours. She had thirty tons of coal on board, and was drawing 7 feet 4 inches (aft) at the start. Six runs were first made on the measured mile to determine the number of revolutions necessary to give a speed of 13 knots; this was found to be 170 per minute, or 785 revolutions per knot. Two of the three boilers were used with natural draft.

The fires were kept light, and a good deal of attention was given to their condition. The steam regulating valves were set to maintain a pressure of 180 pounds in the boilers, and the engines were linked up as high as possible. Observations and diagrams were taken every half hour.

The coal was carefully weighed on this trial, an account being kept of that used. During the 12 hours 9,196 pounds were burnt. The indicator cards, about 240 in number, were worked out afterwards, giving an average of 499.1 indicated horse-power. This comes out 1.53 pounds per indicated horse-power per hour. The fans were not running, but the steam steering engine and evaporator were in use all the time, about three-fourths of a ton of water being distilled. There was more water in the boilers at the finish than at the start.

The *Ardent* had a commissioning trial on the 7th of May, during which she made two runs on the mile, the result being 24 knots and 3,670 I.H.P.

Rocket, Shark and Surly.—Three torpedo boat destroyers, built by Messrs. J. & G. Thomson, Clydebank, have recently completed their trials. The following are their dimensions:

Length on water line, feet	200
Breadth, feet	19.5
Draught, mean, feet	5.25
Displacement, tons, about	235

They have straight stems without bow torpedo tubes, the tubes being mounted on deck instead, giving greater accommodation for the crew. The deadwood is cut away so as to give a free flow of water to the propellers, and to facilitate manœuvring.

The engines are of the triple-expansion type with three cylinders each, the diameters of which are $18\frac{1}{8}$, $26\frac{3}{8}$ and $40\frac{1}{8}$ inches,

and the stroke 18 inches. The steam valves are of the piston type worked by the Stephenson link motion. The screws are of manganese bronze. The boilers are of the Normand type, four in number, and contain 163 square feet of grate and 8,600 of heating surface. The tubes nearest the fire are of steel, and the others of copper. (This change was made after the accident on board the *Shark*, mentioned in the last number of the JOURNAL, and similar changes from copper to steel tubes are now being made in all the "destroyer" boilers which have copper tubes.) There are three smoke-pipes.

The following are the results of the measured mile runs:

	Rocket.	*Shark.*	*Surly.*
Date of trial	Feb. 27.	March 1.	March 28.
Steam pressure	188	184	...
Air pressure	3·5	3	...
Revolutions per minute	398	398.5	398
Speed	27.7	27.5	27.6

The three-hour runs gave—

Speed	27 4	27.6	28.05
Revolutions per minute	396	401	405
I.H.P.	4,200	4,250	...

Banshee, Contest and Dragon.—Vessels of the same general type, built by Laird Brothers, Birkenhead. They are similar in construction and general appearance to the *Ferret* and *Lynx*, described in the last volume of the JOURNAL, and are of the following dimensions:

Length on water line, feet	210
Breadth, feet	19.5
Draught, mean, feet	5.7
Displacement, tons	265

The engines are of the triple-expansion type, with cylinders 19, 29 and 43 inches in diameter and 16 inches stroke, and are both worked from a platform at the forward end of the engine room. There are four boilers of the Normand type similar to those in the *Ferret*.

Their armament comprises one 12-pounder and five 6-pounder rapid fire guns, and two 18-inch torpedo tubes mounted on deck;

the same armament is carried by all the larger boats of this type.

The results of the official trials were:

	Banshee.	*Contest.*	*Dragon.*
Date of trial	Feb. 21.	Feb. 22.	May 7.
Speed, measured mile	27.96	...	27.4
Revolutions, per minute, three hours	345	350	353.
Speed for three hours	27.6	27.4	27.14

Sturgeon.—One of the same class building by the Naval Construction and Armaments Company, Barrow, has had considerable difficulty on her trials. While making her preliminary trials early in the year several tubes in her boilers burst or were burned, resulting in the fatal scalding of some of the crew, which led to the substitution of steel tubes for the copper ones. It is further stated that, in order to get the men to run the official trials of these boats, the company insures their lives for a certain sum for the benefit of their families, and supports the men if injured during the trials.

The *Sturgeon* on an unofficial trial made 27.86 knots with 401 revolutions of her engines, and is said to have subsequently made as much as 29 knots for two hours. On her official trial, which took place on the 11th of April, one of the H.P. piston rods bent, necessitating the discontinuance of the trial. This was the second time that the same rod had bent and interfered with her trials. On her next official trial, April 18, she had to stop for two hours on account of hot bearings; and then, after being under way a short while, the same piston rod, which had given trouble on the former trials, was bent, and the trial had to be again postponed.

The contract price of this boat is £33,977.

Janus and Lightning.—Two of the destroyers, building by Palmer's Shipbuilding and Iron Company, were recently launched from their works at Yarrow. Their dimensions are:

Length on water line, feet	200
Breadth, feet	19.75
Draught, mean, feet	6.5
Displacement, tons	277

Their engines are of the triple-expansion type, both in the same compartment, and are supplied with steam from four Reed boilers placed in two compartments. These boilers are similar in general design to the well known Thornycroft boilers with an upper central steam and water drum and two lower wing water drums, one on each side of the grate. The tubes are arranged very much the same as those of the Thornycroft boiler, except that some ingenuity has been displayed in the form of bends in the tubes, and that the tubes deliver into the upper drum below the water line. There are outside downcast pipes at each end, and removable plates on the lower drums for reaching the tube ends.

The price of these boats is £37,107 each.

Handy.—One of the torpedo boat destroyers building by the Fairfield Company, was launched on the 9th of March. She is 194 feet long, 19 feet 4½ inches wide, has a mean draught of 5 feet 7 inches, and a displacement of about 225 tons. She will have triple-expansion engines and Thornycroft boilers.

Another one of the same type, the *Hart*, building by the same firm, will have Babcock and Wilcox boilers.

Havock and Hornet.—In a letter in the London "Engineer" of April 12, 1895, on the subject of Steamship Resistance, Colonel R. de Villamil gives the following figures for the progressive trial speed and power of the *Havock* and *Hornet*, stating that the figures are from a speed curve furnished by Messrs. Yarrow and Co.:

Speed.	*I.H.P.*	*Speed.*	*I.H.P.*
11	165	20	1,930
12	325	21	2,155
13	500	22	2,390
14	675	23	2,632
15	865	24	2,875
16	1,065	25	3,125
17	1,270	26	3,395
18	1,485	27	3,660
19	1,700	28	3,935

The boilers have been taken out of the *Hornet*, and the copper tubes replaced with steel ones. The cost of these boats was £34,400 each.

Renown.—This first-class sheathed battleship was launched at Devonport on the 8th of May. Like the *Centurion* and *Barfleur* she has been designed to pass through the Suez Canal, and to keep the sea for long periods without docking. Her principal dimensions are:

Length between perpendiculars, feet	380
Breadth, extreme	72.33
Load draught, feet	26.75
Displacement, tons	12,350

She is of the central citadel type, the sides of the citadel being constructed of two strakes of Harveyized steel, the lower 8 and the upper 6 inches thick. The protective deck within the limits of the citadel is 2 inches thick on the flat and 3 on the slope, and forward and abaft the citadel 2 inches thick.

The battery comprises four 10-inch guns in redoubts at the forward and after ends of the citadel, ten 6-inch in casemates on the upper and main deck, ten 12-pounders, twelve 3-pounders, and seven Maxim guns, and five torpedo tubes, four of which are submerged. The armor on the redoubts is 10 inches thick.

The machinery, built by Maudsley, Sons & Field, is similar to that of the other first-class battleships, with cylinders 40, 59 and 88 inches in diameter by 51 inches stroke. Steam is furnished by eight single-ended boilers working at a pressure of 155 pounds. The I.H.P. at natural draft is 10,000, and with forced draft 12,000, the corresponding speed being respectively 17 and 18 knots.

Talbot.—A second class sheathed cruizer, of about the same size as our *Olympia*, was launched at Devonport on the 25th of April. Her dimensions are:

Length between perpendiculars, feet	350
Breadth, extreme, feet	53.5
Draught, forward, feet	19.5
aft, feet	21.5
Displacement, tons	5,600
Coal stowage, tons	1,076

Her engines, which are similar to those of the other vessels of this type, described elsewhere in this number, are of 9,600 I.H.P., with which she is expected to make 19.5 knots. The cylinders are 33, 49 and 74 inches diameter by 39 inches stroke, and the revolutions at full power are 140.

The battery comprises five 6-inch, six 4.7-inch, eight 12-pounders and one 3-pounder, one 12-pounder field gun, four 0.45-inch Maxims, and ten 18-inch torpedoes.

Phœnix.—One of the two twin-screw sheathed sloops, described on page 615 of the last volume, was launched on the 25th of April, at Devonport.

Her length is 185 feet; beam, 32.5; draught, forward, 11, and aft 11.5 feet; displacement, 1,050 tons. She has bunker capacity for 160 tons of coal, and has been designed for 14 knots at full power and for 12 knots with natural draft. Her complement is 106 officers and men.

Her engines are of the vertical triple-expansion type, of 1,400 I.H.P., and have cylinders 13½, 20½ and 31 inches diameter, designed to run at 200 revolutions per minute at full power.

Her armament is six 4-inch and four 3-pounders, and three 0.45 inch Maxims.

Alert.—This sheathed sloop, which was described on page 203 of the last volume, recently completed her natural draft trial with satisfactory results. The result of the eight hours' run was: Revolutions, 178; I H.P., 1,156; speed, 13 knots. The forced draft trial gave 13.42 knots with 1,484 I.H.P., the revolutions being 202.8, and the steam pressure 150.8.

FRANCE.

Hoche.—Owing to the general top-heavy condition of this battleship, extensive alterations have been carried out on board her with a view to increasing her stability. The after military mast has been removed altogether, as also the light shelter deck forward and the admiral's balcony aft. The citadel above the 27-cm. guns has been greatly modified with a view to reducing its weight, six of the 14-cm. guns taken off altogether, and four 65-mm. added. The total saving of weight effected in the remodeling

is said to be as much as 288 tons. Since the completion of the above mentioned alterations, she has had a trial, during which the speed made was 15 knots.

Dupuy de Lôme.—This triple-screw cruiser, whose trials have extended over a period of two years and been repeatedly interrupted by defects of various kinds in the machinery, the last of which involved the removal of the boilers, the cutting out of the furnaces (51 inches in diameter), and the substitution of smaller ones, has finally completed her official trials, the duration of the final one at full power having been reduced to one hour on account of fear lest the boilers might not stand the forcing.

The full speed trial was made on the 2d of April, when she made 19.8 knots on about 13,000 I.H.P. The revolutions of the engines were: Center, 135; starboard, 138; port, 140. The maximum speed reached during the trial was 20.4 knots.

The *Dupuy de Lôme* is of the following dimensions: Length, 374 feet; beam, 51.5; mean draught, 23.2; displacement, 6,198 tons.

Latouche-Tréville.—First class cruiser, mentioned on page 172 of the last number, has completed her trials. On the 4th of February she made an endurance trial at 10 knots and 2,400 I.H.P. On her official full speed trial she made 18.16 knots and 8,450 I.H.P., which was below the design, 19 knots and 9,000 I.H.P., but was regarded as satisfactory. On the 4th of March she made another trial at about two-thirds power lasting 24 hours, when, with 110 revolutions of the engines, she made 16.84 knots and 5,250 I.H.P.

Friant.—This cruiser, a description of which was given on page 171 of the last number of the JOURNAL, has completed her official trials, having made 18.8 knots for two hours with 9,503 I.H.P. On the measured base she made as much as 19.3.

On account of her lack of stability, her two military masts have been cut down to mere poles, four torpedo tubes and the proportionate number of torpedoes removed, and other modifications made in order to improve her condition.

Chasseloup-Laubat.—The trial of this cruiser, a sister vessel to the *Friant*, was carried out on the 28th of February, the speed on the measured base being 18.77 knots with 125.7 revolutions of

the engines. The designed speed was 19 knots with 9,000 I.H.P.

Cassini.—This torpedo cruiser, building by Normand and Company, at Havre, had preliminary trials on the 7th and 18th of March, during the latter of which her engines were run up to 150 revolutions, giving the vessel a speed of 20 knots, when some of the bearings heated and the boilers gave trouble, to remedy which will involve a delay of a month or more.

Lansquenet.—This torpedo boat, which was described in the last volume of the JOURNAL, has, after many modifications, again been making trials. On a preliminary one she made 24.5 knots without difficulty, but while running the official trial she broke a cylinder and piston. She had made 25.45 knots against the tide during one run on the measured base. Her natural draft trial gave 18.79 knots.

Flibustier.—A Normand sea-going torpedo boat has had a preliminary trial, making 25½ knots. She is of 120 tons displacement, and has twin-screw triple-expansion engines and two Normand boilers.

No. 192, engined by Creusot, made 23.54 knots on trial.

No. 73 broke a cylinder and piston while making a trial.

GERMANY.

Wörth.—The following particulars of the machinery of this battleship, described on page 808 of the last volume of the JOURNAL, are from a paper by Professor Carl Busley in "Zeitschrift des Vereines Deutscher Ingenieure."

The engines are of the three-cylinder triple-expansion type, each in a separate water-tight compartment, and work with a boiler pressure of 170 pounds. The main steam valves are double ported piston valves of the Trick type, and are worked by Marshall gear. There are twelve single-ended cylindrical boilers, each with three corrugated furnaces 36 inches internal and 40 inches external diameter. They are 13 feet in diameter and 9.45 feet long, and have an aggregate grate and heating surface of 755.6 and 25,381 square feet respectively. The grates are 6.56 feet long.

The screws are three bladed, 16.73 feet diameter, and have a mean pitch measured at $\frac{7}{10}$ the radius of 17.71 feet.

Her trials, which were made at her designed draught, consisted of two of 24 hours' duration, on the first of which the I.H.P. was 2,143 and the revolutions 66.75; and on the latter 4,144 I.H.P. and 84.4 revolutions, with an air pressure of less than $\frac{1}{4}$-inch in the fire rooms. These were followed by the forced draft trial, which was of six hours' duration, during which the revolutions averaged 109.5, and the I.H.P. 10,088. Subsequently, she made a number of runs on a measured base two miles in length, at a speed of 111 revolutions, which gave a mean speed of 16.6 knots in water of 11 fathoms depth, and as high as 17.2 in 32 fathoms. A week later, she made another 24 hours' trial, and developed 7,945 I.H.P. with $\frac{4}{10}$-inch air pressure.

The machinery of the *Wörth* was built by the Germania-Werft, Kiel.

Hohenzollern.—This vessel, which has been designed as an imperial yacht in time of peace, and for a dispatch vessel in time of war, has been described and illustrated in the same publication by Prof. Busley, from which the following data are taken:

Length over all, feet	400.3
between perpendiculars, feet	382.5
Breadth extreme, feet	45.9
Draught forward, feet	17.3
aft, feet	19.3
Displacement, tons	4,160.

The engines are of the twin-screw triple-expansion type, with cylinders 35.8, 57.9 and 92.5 inches diameter by 37.4 inches stroke, and work with a boiler pressure of 170 pounds. The cylinders are not jacketed. The valves are of the piston type, all worked by the Klug gear. Each engine has two air pumps 19.7 inches diameter and 15 inches stroke, one worked from the I.P. and the other from the L.P. crosshead. The screws are four-bladed, 14.76 feet diameter and 22.64 feet mean pitch, and have a projected area of 41.12 square feet. The crank shafts and pins are $14\frac{1}{2}$ inches diameter and the propeller shafts 15, and the casing of the latter one inch thick. There are two boiler rooms,

each containing two double and two single-ended boilers, all 15 feet in diameter, and the double-ended ones 20.17 feet long. There are four ribbed furnaces at each end, 39 inches diameter, with a combustion chamber 22 inches deep, common to each pair of adjacent furnaces. The tubes are 7.5 feet long between plates, all 3 inches in diameter, the plain tubes .137 inch and the stay tubes .236 inch thick. The front tube sheet is $\frac{7}{8}$ and the back one full $\frac{3}{4}$ inch thick. The total grate and heating surfaces are, respectively, 1,033 and 28,837 square feet.

On her official trial, on a displacement of 4,093 tons, she ran for six hours, the engines making 107 revolutions and developing 9,503 I.H.P. Twelve days afterwards, she made her runs on the measured base to determine the speed at the mean number of revolutions made on the six hours' trial; with 110 revolutions, the mean speed was 21.53 knots, and the maximum during any of the runs 21.82. This is equivalent to a mean speed of 20.94 knots for the six hours; or, considering the method of trial, fully 21. After the six hours' trial, the vessel is brought to her load displacement, and runs made on the base at practically the mean revolutions of the trial, and the speed for the trial determined from the revolutions.

The machinery of the *Hohenzollern* was built by the Vulcan Works, Stettin.

Professor Busley calls special attention to the remarkably spacious fire rooms in this vessel, and in connection with the subject states that after his first visit to the fire room, the Emperor asked him how he found things down there; he expressed his astonishment at finding so much room, whereupon the Emperor replied: "The fire rooms received my personal attention. When I am up here on deck enjoying myself I do not want to feel that the firemen down below are nearly dying with the heat. As it is now, they will have a thoroughly comfortable place to work in even in warm weather." The results will undoubtedly justify the sacrifice that must have been made to accomplish this result, and the end sought in her construction, the maintenance of her speed for an indefinite period, accomplished.

RUSSIA.

Rurik.—The following details of this armored cruiser are mostly from "Mittheilungen aus dem Gebiete des Seewesens:"

Length between perpendiculars, feet	412
on water line, feet	425.8
over all, feet	434.7
Breadth, extreme, feet	67
Draught, mean, feet	25.6
Displacement, tons	10,760
Coal at normal draught, tons	1,634
total in bunkers, tons	1,968

She has an armor belt extending about $\frac{8}{10}$ her length; it is 6.9 feet deep, 9.8 inches thick at and above the water line, and 5 inches below. Her protective deck has a maximum thickness of 2.5 inches, tapering to 2 inches forward and aft. Her battery comprises four 8-inch, sixteen 6-inch, fourteen 5-inch, and eighteen small rapid-fire guns, besides five torpedo tubes.

She has two masts with double military tops, and three smoke pipes.

The original design of machinery contemplated two sets of triple-expansion engines of 6,000 I.H.P. each; but this was subsequently changed to four engines, two for each screw, with a collective I.H.P. of 13,070 I.H.P. with natural draft. The boiler installation has also been changed, so that instead of all double-ended, she now has one-half double-ended and one-half Belleville boilers. The former, of which there are eight, are six-furnace boilers, and are placed in four water-tight compartments.

The trials were made on the 15th, 20th and 23d of October, 1894, at a mean draught of 24.12 feet and a displacement of 9,407 tons, and resulted as follows:

Runs	*First.*	*Second.*	*Third.*	*Fourth.*
Revolutions per minute	80.5	82	82	82
Speed, knots	18.04	18.97	18.74	19.17

The maximum I.H.P. on these trials, which were at natural draft, was 13,370.

A sister ship, the *Rossija*, now building, is to have triple-screw

engines, the side ones of 7,250 I.H.P. each, and the center one of 2,500.

Standard.—The Russian imperial yacht, building by Burmeister and Wain, Copenhagen, was launched from their works on the 10th of March. The following are her dimensions:

Length on water line, feet	370.64
Breadth, feet	50 34
Draught, mean, feet	18.53
Displacement, tons	5,116

The engines will be twin-screw, triple-expansion of 10,600 I.H.P., which is expected to give her a speed of 20 knots. Belleville boilers will be fitted. The screws are of Delta metal, 16 feet in diameter and 27 feet pitch, with a blade area of 77 square feet for each screw.

The yacht will, on the upper deck, have two saloons, a drawing-room, and a dining saloon capable of seating sixty persons. On the main deck will be a saloon for the Imperial family furthest aft, besides a smaller dining saloon and rooms for the Emperor and Empress, the Dowager Empress, and the grand dukes. There will also be rooms for the Imperial suite, the Naval Minister, and for the General Admiral. Forward will be the quarters of the officers. On the intermediate deck will be dining saloons and saloons for the suite, and some rooms for servants and crew. On the lower deck will be rooms for the non-commissioned officers, etc.

SPAIN.

Carlos V.—This armored cruiser was launched at Cadiz on the 12th of March. She is somewhat similar in general design to the English cruiser *Blake*, and is of the following dimensions:

Length on water line, feet	380
over all, feet	404.75
Breadth, feet	67
Depth, feet	39.75
Draught, mean, feet	24 54
Displacement, tons	8,710

She was designed for a speed of 19 knots with natural and 20 with forced draft, the I.H.P. being respectively 15,000 and 18,500 I.H.P. The bunker capacity is 1,770 tons.

There is a thin armor belt, in two thicknesses of one inch each, for a length of 167 feet, terminating forward and aft in bulkheads 6 feet high, and a protective deck 2 inches thick. The battery comprises two 11-inch guns in barbette turrets, one forward the other aft, protected by 9.8-inch plates, and the covering plate 4 inches thick; eight 14-cm., four 10-cm., and two 7-cm. rapid-fire guns, and eight machine guns.

The engines are vertical, twin-screw, triple-expansion with cylinders 52, 77.2, and two of 82.1 inches diameter by 45.3 inches stroke, working with a boiler pressure of 150 pounds per square inch. There are twelve single-ended boilers 16.3 feet in diameter and 9.85 feet long. The screws are four bladed.

Quiros.—A small single-screw gunboat, built for service in the Phillipine Islands, was launched from the Kowloon Docks, Hong Kong, on the 24th of January. Her dimensions are:

Length over all, feet	145.25
Breadth, feet	22.83
Depth, feet	11.
Displacement, tons	347.

The engines, of 500 I.H.P., are triple-expansion, with cylinders 13, 21 and 35 inches diameter and 24 inches stroke, and work with a boiler pressure of 170 pounds, steam being supplied by two single-ended boilers. The speed expected is 12 knots.

The battery consists of two 57-mm. Hotchkiss and two 5-barrel Nordenfelt guns.

MERCHANT STEAMERS.

The "Marine Journal" and "Marine Review" say that the Red D Line has entered into a contract with the William Cramp and Sons' Ship and Engine Building Co. for the construction of a freight and passenger steamer for the trade between New York and Maracaibo. The dimensions are :

Length over all, feet	256
on water line, feet	248
Breadth, molded, feet	38
Depth, molded, feet	17.4
Draught, mean, feet	10
Gross tonnage	1,500

Her speed is intended to be 11 knots in service and 12 knots on the four hours' trial under the Postal Subsidy Act, which will place her in the fourth class of the Reserve.

The hull is to be of steel. There will be a steel deck for half the length of the vessel amidships, a steel deck house containing limited passenger accommodations, a forecastle for the crew forward, and turtleback aft. The vessel is divided by five watertight bulkheads. There is a water ballast compartment aft of the engines and one at each end of the vessel, which will take about 200 tons of water ballast. The number of first class passengers provided for is twenty-four.

She has three pole masts with leg of mutton sails.

Her engines are of the triple-expansion type, of 1,000 I.H.P., the cylinders being 18, 28 and 45 inches diameter by 30 inches stroke, steam for which will be supplied from two boilers working at 160 pounds pressure. There is also a donkey boiler on deck. The pumps will be independent.

Pomona.—The following details of two trials of the machinery of this steamer, one made before and the other after fitting the Howden system of forced draft to the boilers, is given in

"Industry," on the authority of Mr. G. W. Dickie, of the Union Iron Works:

In the first trial the cylinder diameters were 23, 34 and 56 inches, and in the second $23\frac{5}{16}$, $34\frac{3}{16}$ and 56. The stroke is 46 inches. It thus appears that the cylinders were re-bored before the second trial. The heating surface was 2,963 square feet, the grate surface before the change 102, and after 73 square feet.

S. S. Pomona.—Result of machinery trials before and after fitting with Howden's forced draft system.	VOYAGES. San Francisco to Eureka, Aug. 29, 1894. without forced draft.	VOYAGES. San Francisco to Port Harford Nov. 29, 1894. with forced draft.
Duration of trial	6 hours.	16 hours.
Steam pressure, engine room	145.4	147.8
first receiver	41	42
second receiver	5.59	6 15
Vacuum	25.4	24.8
Revolutions per minute	91.22	99
Mean pressure in cylinder, H.P.	51.1	62
I.P.	21.23	25.8
L.P.	8.71	10.8
M.P. reduced to L.P. cylinder	25.06	31.5
Average indicated horse power, main engines	1,020	1,388
auxiliaries	58	84
total	1,078	1,472
Air pressure in inches of water	.29	.77
Coal burned, total	14,415	49,410
per hour	2,402.5	3,088
per I.H.P. per hour	2.22	2.09
per sq. ft. of grate surface per hour	23.5	42.3
Heating surface per I.H.P., square feet	2.74	2.00
I.H.P. per square foot of grate	10 56	20.16
Grate surface in use, square feet	102	73
Coal (Franklin), quality	Good.	Bad.
Draught of vessel, forward, feet and inches	7–11	10–8
aft, "	12–6	13–6
mean, "	10–2½	12–1
Displacement, tons	1,150	1,455

Prinz Heinrich and Prinz Regent Luitpold.—A short notice of these twin-screw steamers, built by Schichau for the Australian trade of the North German Lloyd, was given in the last volume of the JOURNAL, but as additional data of them are now at hand, all the available data are given here. The dimensions are:

Length on water line, feet	450
Breadth, feet	51
Depth amidships, feet	33
Gross register tonnage	6,100
Load draught, feet	26
Displacement, load, tons	11,750

She has a double bottom and is divided into ten water-tight compartments. There are three decks from end to end of the ship, and an orlop deck, so that extensive room is provided for cargo, there being five compartments—two large ones forward and three aft—for cargo in the hold and orlop deck, while the next deck is reserved for cargo or for emigrants. There is, too, a large forecastle, with long promenade deck amidships, the full width of the ship, and a poop, the three being connected by gangways over the wells, providing access to the hatchways forward and aft.

The accommodations for passengers are very good, everything being in first-class style. She can carry 75 first-class, 76 second-class and 960 steerage passengers.

The two engines are in the same compartment, and are similar in general design and arrangement to those supplied the Austrian battleship *Tegethoff*. The working platform is at the forward end. They are of the triple-expansion type, with cylinders 28.35, 46 and 70.86 inches in diameter by 47.25 inches stroke, and work with a boiler pressure of 175 pounds. Piston valves are used on the high pressure, and slide valves for the other cylinders, all operated by Stephenson link motion. The condensers are cylindrical, of galvanized iron, and have a combined cooling surface of 7,330 square feet. The air pumps are worked from the low-pressure crossheads. The shafting throughout is of steel, the total length for each engine being 170 feet, and that of the propeller shaft 50.84. The crank shafts are in interchangeable sections. The propellers are 15 feet in diameter, the hubs of cast steel and the blades of bronze; the pitch is 19.68 feet.

There are two single and two double-ended boilers, each 14.9 feet in diameter, the former 9.18 and the latter 17.7 feet long; the grate surface is 378, and the heating surface 16,512 square feet.

The furnaces are Purves' ribbed, 43.3 inches diameter. There is also a donkey boiler on the upper deck.

The following are the data of the official trial:

Speed, knots	15.5
Displacement about, tons	9,000.
Steam pressure	175.
Vacuum	27.2
Revolutions per minute	87.5
I.H.P., starboard	2,760.5
port	2,805.
collective	5,565.5

On the first voyage of the *Prinz Regent Luitpold* to Australia her average speed was 13.55 knots, the revolutions of the engines 80, and the ship's draught 22 feet. The I.H.P. was about 4,670.

Columbia and Alma.—Short descriptions of these vessels were given in the last volume of the JOURNAL, and also on page 191 of the last number. Since that, however, more complete description has appeared in "Engineering," from which this is taken.

They were built by Messrs. J. and G. Thomson from specifications prepared by Professor J. Harvard Biles, and are of the following dimensions:

Length on water line, feet	270
Breadth, feet	34
Depth, feet	23
Gross tonnage	1,145
Net tonnage	410

The frames are of the reverse angle type, and spaced about 21 inches apart. The interior is subdivided by 11 water-tight bulkheads into 12 compartments. The bulkheads are constructed of plates of flanges vertically instead of angles, the plates themselves overlapping—a system of construction which resists greater lateral pressure than the usual method of angle stiffeners. Three of the compartments are occupied by machinery, one in the center of the ship by first-class state rooms, one at the after end by second-class accommodation; while two at each end are entirely devoted to cargo, the others forward and aft being for storage, cargo, &c. The vessels would float with any two adjoining compartments flooded.

The center compartment in the ship, 33 feet long, was entirely given over to cabins arranged on three decks. Thus in the bridge house on the promenade deck there are five separate rooms on either side of a corridor, with a number of other rooms forward, principally for officers. On the next deck there are rooms at the side of the ship, with a number of rooms again in the center, with smoking room aft and breakfast and supper room forward, while on the main deck there are 16 rooms in the center.

Thus there are berths for 104 first-class passengers in some 43 separate rooms. The state-rooms situated on the promenade deck are usually for two passengers only, and are specially light and airy, having large square ports, while those on the upper and main decks are exceptionally lofty, and throughout the greatest care has been taken to thoroughly ventilate the cabins. From the different compartments there are led air trunks, by means of which vitiated air is exhausted into annular spaces round the funnels, while fresh air is suppled by large cowls, and is led through trunks of large sectional area to the different state rooms, the supply to each of which is regulated by a sliding louvre. The first class dining saloon is situated on the upper deck, immediately forward of the machinery, and is beautifully fitted up, being panelled in polished sycamore, having carved panels alternating with mirrors. On the promenade deck is a spacious smoking room panelled in dark oak, furnished with marble top tables and with couches upholstered in red morocco.

There are two four-cylinder triple-expansion engines, in the same water-tight compartments, the working platform at the forward end, with cylinders 19, 29 and 33½ inches in diameter by 30 inches stroke. The valves are of the piston type with balance cylinders, and are worked by Stephenson link motion. The cylinders are supported at the back on cast columns, but the front columns are of forged steel, for securing both accessibility and lightness. The crankshafts are in two pieces, and, together with the thrust tunnel and propeller shafts, are of steel. The thrust blocks are cast iron, and the four collars are of cast steel. The latter are lined with white metal, and are of the horseshoe

type, each separately adjustable. The screw propellers have three blades of manganese bronze fitted to bosses of cast steel.

The surface condensers are placed in the wings of the ship. They are built up of brass sheets, and the water spaces and casing doors, together with main eduction and other branches, are all gun-metal castings. The condensing water to each condenser is circulated by a centrifugal pump driven by an independent open-fronted engine. The air pumps are bolted to the backs of the soleplates in the position shown on the engravings, and are driven by levers off the forward low pressure cylinder.

In addition to the engines mentioned, there are in the engine room: One set electric light machinery; two of Weir's feed pumps and heater; two duplex bilge and fire pumps; one duplex sanitary pump, and one duplex fresh water pump.

Steam at 160 pounds pressure is supplied by two large single-ended boilers with four furnaces in each, worked under forced draft on the closed stokehold system, the air being supplied by two double-breasted fans, one in each stokehold. These boilers are similar in their arrangement to those fitted by Messrs. Thomson to many fast cruisers.

The guaranteed speed was 18.5 knots, which was easily maintained on a six hours' trial with 184 revolutions of the engines. On the measured mile the following results were obtained from the *Alma*:

	Starboard.		*Port.*
Speed, knots	...	19.3	...
Revolutions	191	...	192
I.H.P	1,910	...	1,830
total	...	3,740.	...
Steam pressure	...	160.	...
Vacuum	...	26.5	...

The high speed, with such comparatively large power, is explained by the fineness of form of the vessel.

Aco.—The results of the engine trials of this tank steamer, built by Messrs. D. J. Dunlop and Co., Glasgow, are from the "The Engineer," (London):

The *Aco* is of the following dimensions: Length between per-

pendiculars, 333 feet; beam, 43 feet; depth, molded, 30 feet; carrying capacity, 4,275 tons cotton seed oil and 600 tons coal.

She has a single-screw triple-expansion engine with cylinders 25, 40½ and 65 inches diameter by 48 inches stroke, designed to work with a boiler pressure of 160 pounds. There are two boilers 13.5 feet diameter and 16.25 feet long, each with six furnaces; the grate surface is 148, and the heating surface 6,128 square feet. The air pump is 22 inches diameter and 24 inches stroke, and the circulating pump 13 inches diameter and 24 inches stroke, the latter double acting, and both worked by the main engine. The condensing surface is 2,557 square feet.

On the trials, at 10 knots, careful measurements were made of the coal and water, and indicator diagrams taken every half hour; the feed water heater was not used, and all feed water passed through a meter before going to the boiler. The condition of the fires was "judged" at the beginning and end of the trials.

The weight of her machinery is given as follows:

	Tons.
Engines complete, including steam and exhaust piping and auxiliary engines	173
Boilers, mountings, funnel, auxiliary boiler, and fittings in fire room	190

THREE HOURS' FULL POWER TRIAL, TWELVE KNOTS.

Steam pressure in boilers	154.3
first receiver	52.6
second receiver	8.07
Vacuum	28.46
Cut off from beginning, H.P	.625
I.P	.6325
L.P	.6195
Nominal rate of expansion	10.81
Same, allowing for clearance	8.45
Revolutions per minute	83.5
I.H.P., H.P	729.54
I.P	741.96
L.P	934.5
total	2,406

FOUR HOURS' TRIAL AT TEN KNOTS.

Steam pressure in boilers	154.3
at engine	152.1
in first receiver	54.6
second receiver	0.33

Vacuum	29.08
Revolutions per minute	67.60
Rate of expansion, nominal	17.8
Same, allowing for clearance	11.84
I.H.P., H.P.	366.11
I.P	456.75
L.P	460.64
total	1,283.5
Temperature of injection	47
discharge	71
feed	52.47
gases 15.5 feet above grate	551.2
Water gauge pressure at same point, inches	$\frac{7}{16}$
Barometer	30
Coal per hour, pounds	2,000.25
per I.H.P., pounds	1.558
per square foot of grate, pounds	13.5
Water per hour, pounds	18,907.5
per I.H.P., pounds	14.72
per pound of coal, pounds	9.45
equivalent per pound of coal from and at 212, pounds	11.48

Sylvania.—The following particulars of this vessel are in addition to those given on page 191 of the last number of the JOURNAL.

There are nine watertight bulkheads extending to the upper deck. In all there are 24 compartments for water ballast, and part of the double bottom under engines may be utilized for carrying reserve fresh water for cattle or boiler use. The entire vessel, including all the holds, 'tween decks, engine and boiler spaces, and cabins, is lighted by electricity on the double-wire system, generated by two compound wound self-regulating dynamos situated in the engine room. At each of the seven hatches there is a cluster of 16 lamps of 16-candle power. The cattle fittings are on the most approved plan, and comprise Mr. Wm. Wyllie's patent slingable fittings, which enable the whole space to be closed and utilized for cargo. Special attention has been given to the safety of the cattle, and the water pipes are so arranged that water may be carried to any of the stalls. Utley's patent cowl and combination vents are liberally distributed throughout the ship. The two sets of triple-

expansion engines have cylinders 22½, 36½ and 60 inches in diameter by 48 inches stroke. There are two main boilers working on Howden's system of forced draft; they are 15 feet in diameter and 20 feet 3 inches long, each with six 43½-inch furnaces, and containing 229 square feet of grate and 9,610 of heating surface. The working pressure is 180 pounds. There is also a large auxiliary boiler for winches, electric light, etc. The maximum speed on trial was 15¾ knots, and the I.H.P. 5,570.

Aberdeen.—The following particulars of the machinery of this vessel, which is fitted with the Fleming and Ferguson water tube boiler, as mentioned on page 642 of the last volume of the JOURNAL, are from the "Marine Review:"

Number of engines	One; quadruple expansion.
Diameter of cylinders, inches	18, 28, 35 and 54
Stroke of piston, inches	38
Clearance, top, inch	½
bottom, inch	⅝
Maximum revolutions per minute	96
Indicated horse power with natural draft	1,510
Steam pressure in boiler, pounds	220
Boilers	Patent "Clyde" water tube.
Number of boilers	2
Diameter of drums	Top, 6 feet; bottom, 3 feet.
Diameter of tubes, outside, inches	2½
Heating surface in one boiler, square feet	1,412
Grate surface in one boiler, square feet	50
Weight of one boiler complete without water, tons	27.69
with water, tons	35.71
Feed pumps	Ordinary type, wrought off air pump levers.
Number of pumps	2
Diameter of pumps, inches	3½
Stroke of pumps, inches	16
Propeller	Built type; four blades; manganese bronze.
diameter, feet	12
pitch, feet	16
surface, square feet	43
Speed of vessel, knots	13

YACHTS.

Giralda.—This yacht, which was described in the last number of the JOURNAL, won the Bennett cup at the races at Nice on the 11th of April, having covered the distance of 26.5 miles in 1 hour 8 minutes and 22 seconds, which, if the distance and time are correct, is equivalent to a speed of 23.25 knots, though it is given as 22.2 and 22.6 knots.

New Alcyone.—A steam yacht designed by Mr. Geo. H. Warrington, Jr., of Chicago, for Mr. H. F. Balch, of Minneapolis, is now under construction on the Lakes. Her dimensions are:

Length over all, feet	135
Breadth, extreme, feet	17
Depth, molded, feet	11
Draught aft, feet	8
Displacement, tons	144

She will be schooner rigged, spreading about 3,000 square feet of canvas, and will stow 32 tons of coal.

The steel plates to be used in the hull will be $\frac{5}{16}$ inch at the garboard and sheer strakes, the rest being $\frac{1}{4}$ inch. Her deck houses will be of steel, covered with mahogany and the interior will be finished in the same woods.

Her engine will be triple-expansion, with cylinders 12, 19 and 32 inches diameter by 18 inches stroke, and her boilers, two in number, of the Warrington water tube type. The boiler room occupies 23 feet lengthwise the vessel and the engine room 18. Her screw will be 6 feet 6 inches in diameter.

The speed guaranteed is 18 miles on trial, and she is expected to make 14 in ordinary cruizing.

The *New Alcyone* will carry an ice machine, and an electric storage battery plant for lighting and cooking purposes, and will be equipped with a powerful searchlight. She will carry an elec-

tric launch 24 feet long, a life boat 22 feet in length, and a dinghy 16 feet long. She is expected to cost $100,000.

The Detroit Boat Works have under construction a single-screw steel yacht for Merrill B. Mills, Esq., of the Detroit Yacht Club, which will be of the following dimensions: Length on water line, 110 feet; length over all, 132 feet; breadth on deck, 17.5 feet; depth, 8.75 feet. She will be rigged as a two-masted schooner, and will have three water-tight bulkheads, one a collision bulkhead, and the others at the ends of the machinery space. Her bunkers stow about 20 tons of coal.

Her engine will have cylinders 10, 16 and 26 inches diameter and 16 inches stroke, and has been designed for 350 I.H.P. Steam will be furnished by two Taylor water tube boilers each 8 feet in width and 9 feet long, both having 1,600 square feet of heating surface. The screw will be 4.5 feet in diameter.

She will be fitted up in most approved style. Forward will be a large deck house finished in red mahogany. It will be 24 feet long and 8 feet wide. This will be used as a dining and lounging room. The ceilings will be electro-plastic metal, and around the sides will be continuous divans, upholstered in the finest morocco leather, in colors to harmonize with the rugs and decorations on the ceiling. The floor will be covered with linoleum and rugs.

On top of the deck house, which will be surrounded by a brass railing, will be the steering wheel, binnacle and a powerful electric search light. Another feature will be an electric flag of 85 colored bulbs, forming the commodore's private colors. From each side of the top of the deck house a bridge runs to the edge of the ship. A mahogany staircase leads down to the smoking room, containing sofas on each side, which can be converted into berths. A door from the smoking room leads to the lavatory and galley. Forward of the galley will be located the crew's quarters, containing eight bunks of galvanized piping. Aft the mainmast is located a companionway. A mahogany staircase leads to the passage below the deck from which admission is gained, on the starboard side, to the owner's stateroom. It will be a marvel of beauty. The interior woodwork of this

room, and the bath room adjoining, will be of Spanish white mahogany, natural finish and in Louis XIV style. The ceilings will be decorated in ivory with raised ornaments of silver. The berth will be finely decorated, the hangings being of the same general style as the deck house, of brocaded silk.

The main saloon is the full width of the ship and 17 feet long. The woodwork is quartered Cuba birch, and the ceiling will be made of raised ornaments, festoons and wreaths, colored and richly gilded, Louis XIII style. There will be continuous divans around the room which can be changed at will into four commodious berths, all upholstered in velour imperator, specially designed and imported for this room. The only occasion on which this material was ever before used in America was in the Pullman cars exhibited at the World's Fair. Leading into the main saloon will be two spare staterooms. The walls will be covered with French cretonne, tufted. The hangings will be light silk, trimmed with lace. There will be davits for two lifeboats, finished in natural wood with mahogany trimmings and gratings, and nickel plated fittings. There will also be a twenty-one foot naphtha launch.

The interior of the boat will be lighted by fifty 16-candle power incandescent lamps furnished by the dynamo in the engine room, in conjunction with a storage battery plant. A feature is the burglar alarm on the after companion way, communicating with an annunciator in the crew's room. [From the "Detroit Free Press."]